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POWER AND THE ENGINEER

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Vol. 29. Index to Advertisers and Classified Index Following Reading Matter. No. 25.

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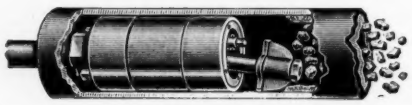
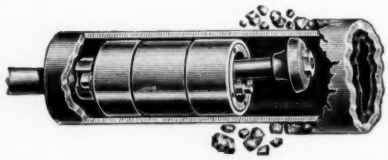


The Dean removing scale from the tube of a water tube boiler.



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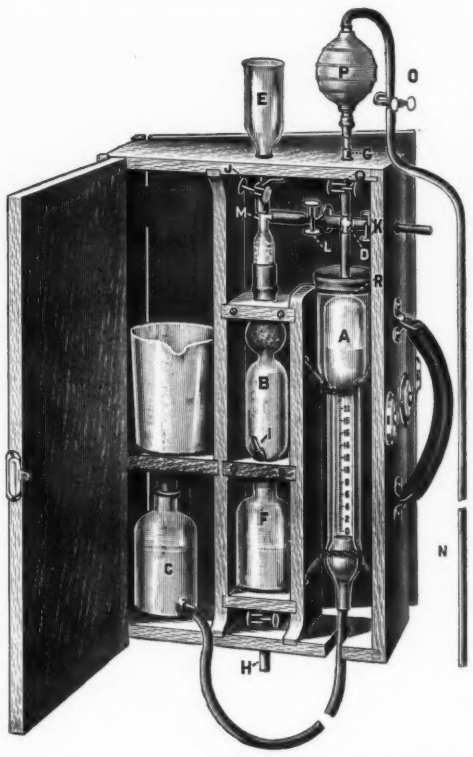

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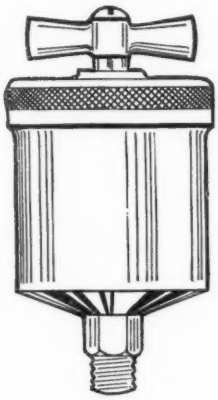

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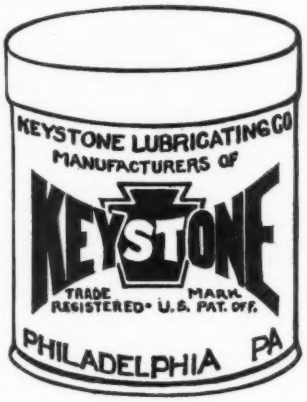

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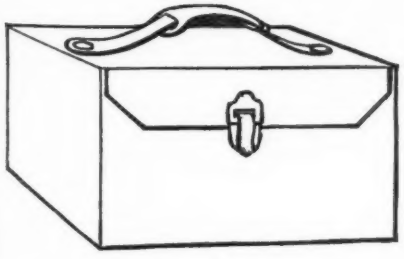

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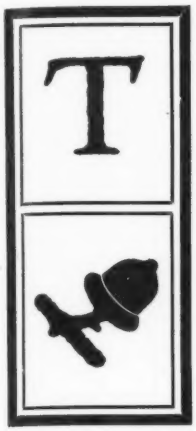

THERE is always one great pioneer and leader in the development of every important industry. Twenty years ago, the fact that great savings could be made in fuel consumption and that boilers and boiler tubes could be made to last much longer, but for the detrimental action of boiler feed waters, interested some young chemists in Chicago in the possibility of surmounting this difficulty by making careful analytical study of different classes of waters, and finding scientific reagents for the elements that produced the trouble.

Thus began the work of the DEARBORN COMPANY. That it has been successful is attested by the fact that thousands of steam plants use their products, prepared according to analysis of the waters, and find absolute relief from the troubles formerly experienced.

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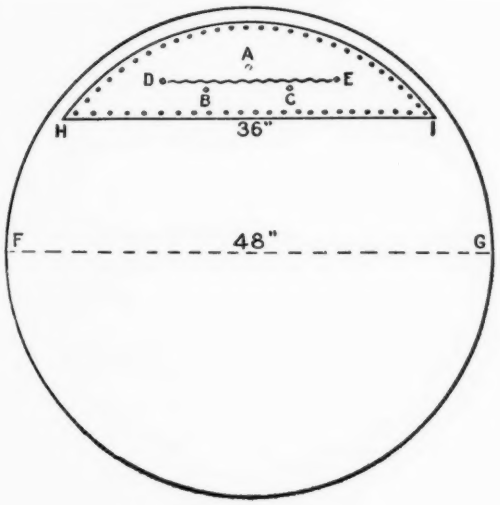

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¶ Prevents foaming and pitting of boilers.

¶ Does away with the deleterious effect of oil or grease in boilers.

¶ Causes no discoloration where live steam is used and does not affect the most delicate fabric it may come in contact with.

¶ It does not affect the sheets or tubes of boilers or interfere with the joints or packing. In fact it stands alone, unequalled by any goods purporting to do the work.

¶ If you are a large user of steam try a barrel of all the different makes of compound on the market, giving them a fair and impartial trial, and then **LET US SEND YOU A BARREL FREE,** compare ours with others and we will receive all your orders in this line thereafter.

¶ Send us a gallon of your boiler feed water or a piece of scale made from the water you use, we will analyse the same free of cost to you and make you the most liberal proposition within the sphere of reason.

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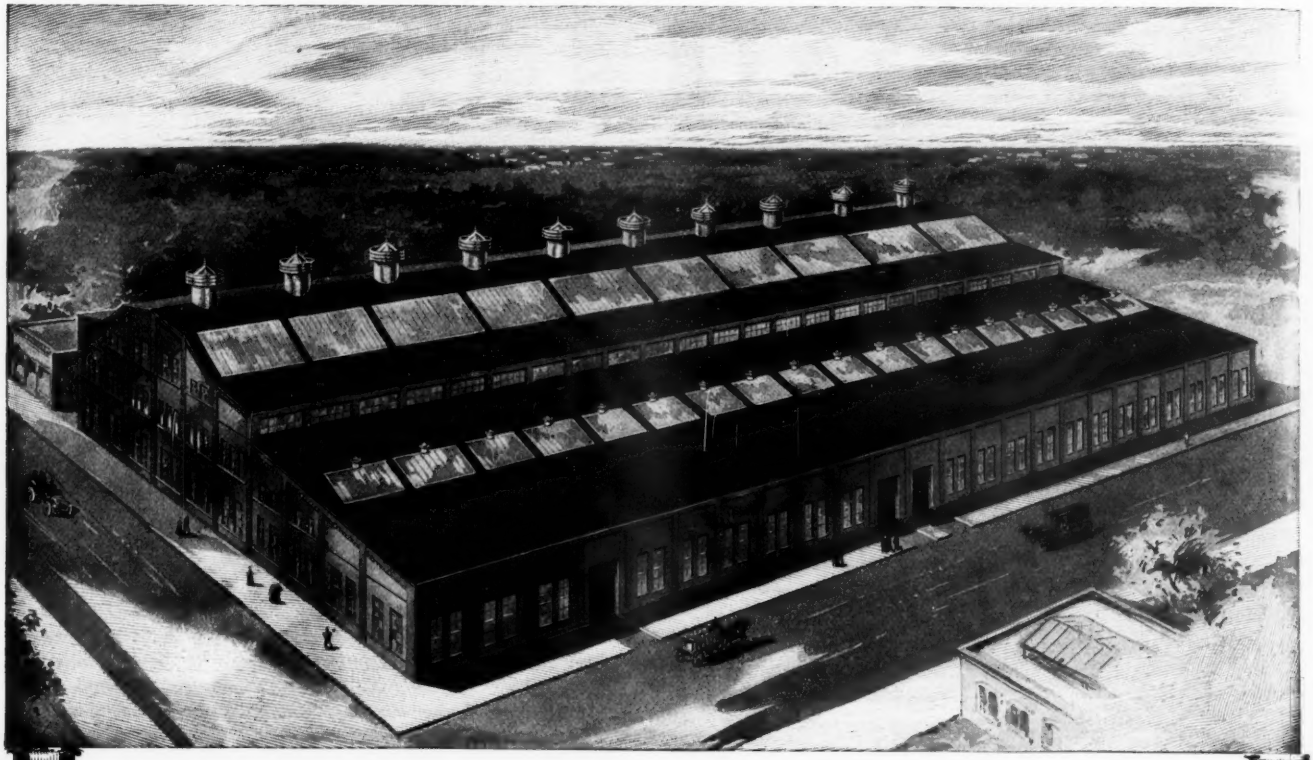

Metal top with sliding-sleeve damper (patented).

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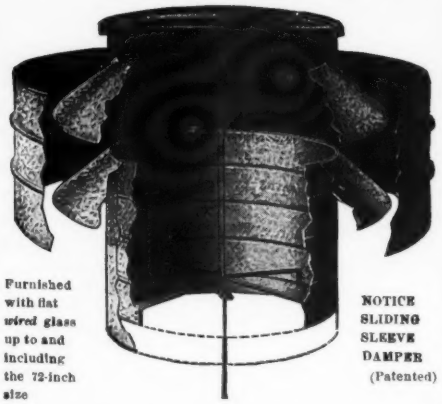
The Jones & Lamson Machine Co. uses 80 "Burts;" The Yale & Towne Mfg. Co. 20; The Florida East Coast Ry., 48. Additional list of users on request.



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A Notable Producer Gas Power Station

First Railway and Lighting Plant in which 60-Cycle Alternators Are Driven in Parallel by Gas Engines on Bituminous Producer Fuel

BY CECIL P. POOLE

It is not a long time since the parallel operation of alternating-current generators driven by gas engines under the most favorable of commercial conditions was considered a dubious proposition by progressive gas-engine builders, while the conservatives would not even consider it in any aspect. Down in the little city of Charlotte, N. C., however, the people are not merely progressive but imbued with a chronic determination to "go you one better" on any industrial, commercial or engineering question, no matter who you are or what you may have achieved; they seem to be born pioneers. Therefore, it is not so remarkable, after all, that the first 60-cycle electric railway and lighting plant in the world to be driven successfully by gas engines supplied from bituminous producers should have been installed in Charlotte. They have lots of other "first time performances" down there, too, but those are irrelevant to the present story.

The Charlotte Consolidated Construction Company, familiarly termed the "Four C's," is operating about 26 miles of double-track electric railway, supplying 200 arc lamps and 12,000 incandescents from its one gas-power station, with no apparatus entirely in reserve, and the service has suffered no interruptions whatever since last June, when the oil pipe leading to the crank-pin box of one of the engines broke off. The damage could not be repaired without shutting down the engine, and it could not be kept running without oiling the crank pin, of course.

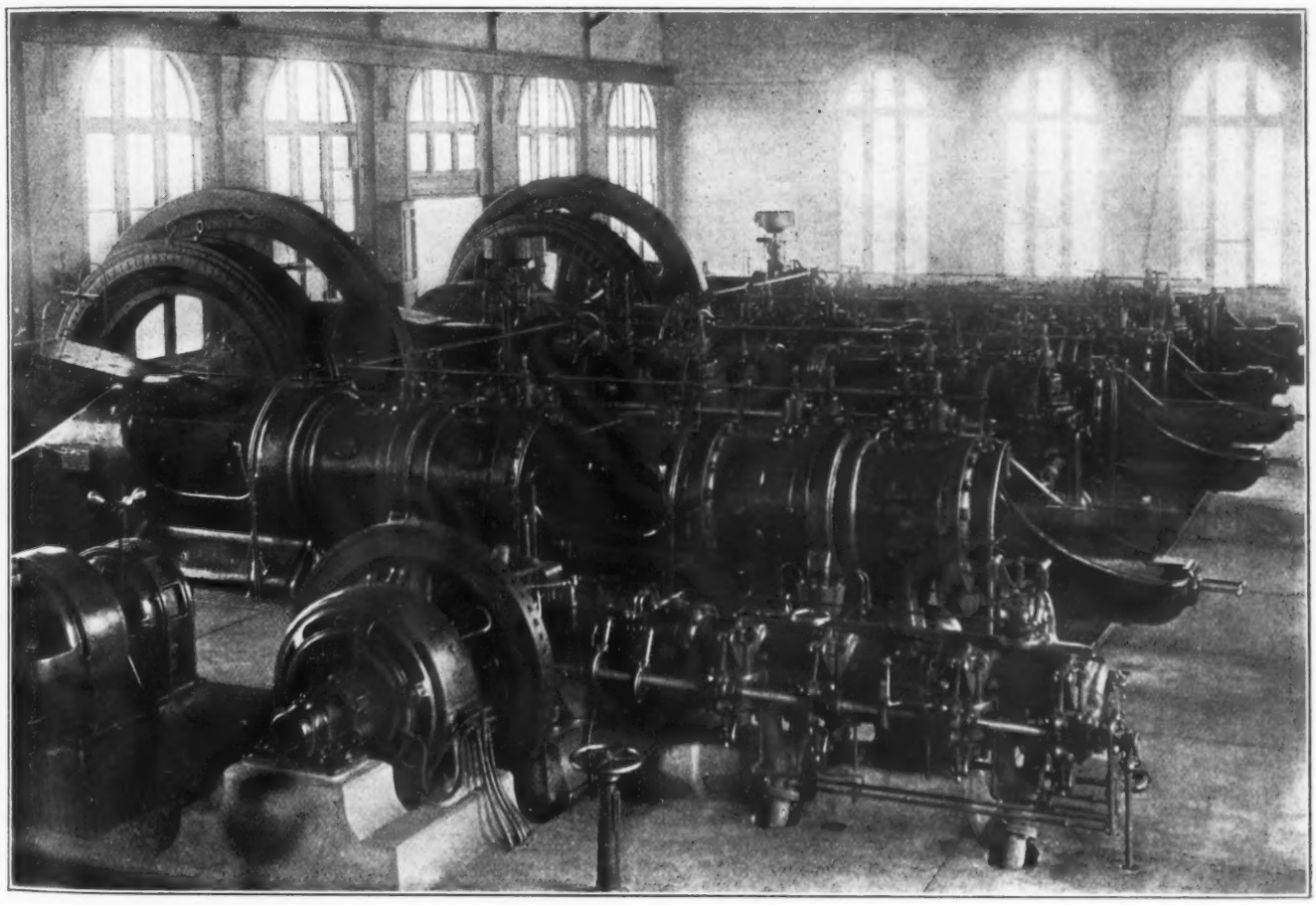


FIG. 1. VIEW OF FOUR C'S POWER HOUSE FROM THE SWITCHBOARD GALLERY

Producer Equipment

The producers are of the well known Loomis-Pettibone down-draft type, built by the Power and Mining Machinery Company. There are two "sets," each comprising two generators, a vaporizer and a wet scrubber; a double dry scrubber, built in a single structure, serves the two sets. Fig. 2 shows the layout of the producer equipment and Fig. 3 shows two sectional elevations at right angles to each other. The generators are each 7 feet in diameter inside the linings, and 14 feet high outside. Each one is provided with cleaning doors located 60 degrees apart around the shell, as indicated in the plan drawing; the two generators of each pair are connected permanently by a 20-inch nozzle at the extreme top of

the shell (see Fig. 3), but each has its own outlet to the vaporizer, as usual. The vaporizer is a simple vertical boiler, 5 feet in diameter and 23 feet high. The working steam pressure is 100 pounds per square inch, gage.

The wet scrubbers are of the usual coke-filled type, 7 feet in diameter and 26 feet high, over all; on top of each one is the auxiliary dry scrubbing compartment characteristic of this make of apparatus. The double dry scrubber is 10 feet in diameter and 5 feet high. The piping of the producer plant is clearly shown by Figs. 2 and 3.

The producers are operated alternately, one set at a time until the vacuum gets up to about 14 inches of water; then the other set is put to work and the dirty set shut down and cleaned. Under present load conditions this means shifting from one set to the other about every ten days.

The company operates also an illuminating-gas plant which is in a building adjoining the power-producer house. The auxiliary boilers shown in the drawings supply steam to that plant and for driving the fan engines in both plants. The gas from the producer plant is led by a 16-inch main to a holder of 60,000 cubic feet capacity located near the power house. From the holder a 20-inch pipe carries the gas to the power house.

The Engines and Electrical Equipment

The power-house equipment at present comprises two twin tandem double-acting Snow engines with General Electric alternators each located between the two frames of its engine; a Type B Snow engine direct-connected to a direct-current exciter dynamo; a 40-kilowatt motor-driven exciter; two rotary converters for "rectifying" the alternating current for the railway system, and a little motor-generator for charging a set of storage batteries which supply ignition current.

The main engines are of exactly the same design as the one at Olean, N. Y., which were fully described in Power and The Engineer for July 14, 1908. The Charlotte engines, however, are 24 inches diameter of bore and 36 inches stroke, and the mixing valve, of course, is designed for producer gas instead of natural gas. The speed is 120 revolutions per minute, giving a piston speed of 720 feet per minute. The engines are equipped with simple mechanically operated make-and-break igniters, which are supplied with current at 10 to 11 volts by a set of "Chloride accumulators. The main generators are three-phase machines of the usual revolving-field type; they deliver current at 2300 volts and 60 cycles, and are rated at 540 kilowatts at 90 per cent. power factor and 40 per cent. temperature rise each.

The motor-driven exciter is the one regularly used; it is a 40-kilowatt 125-volt dynamo, driven at 720 revolutions per minute by a 60-horsepower induction motor which takes current directly from the 2300-volt busbars. The Type B engine cylinders are 12 inches diameter of bore and 16 inches stroke; the exciter it drives is of the same voltage and output ability as the motor-driven machine, but its speed is 260 revolutions per minute. This unit is only for emergency service.

The rotary converters are of the usual type found in railway plants. One is of 300 and the other of 500 kilowatts ability.

The switchboard is located on a gallery at one end of the power house, as shown by Fig. 4. The view shown by Fig. 1 was taken from one end of the switchboard gallery. Fig. 5 is a plan of the power-house equipment and shows the arrangement of the apparatus and piping. Space has been left in the power house for another unit and in the gasometer house for its intake connections.

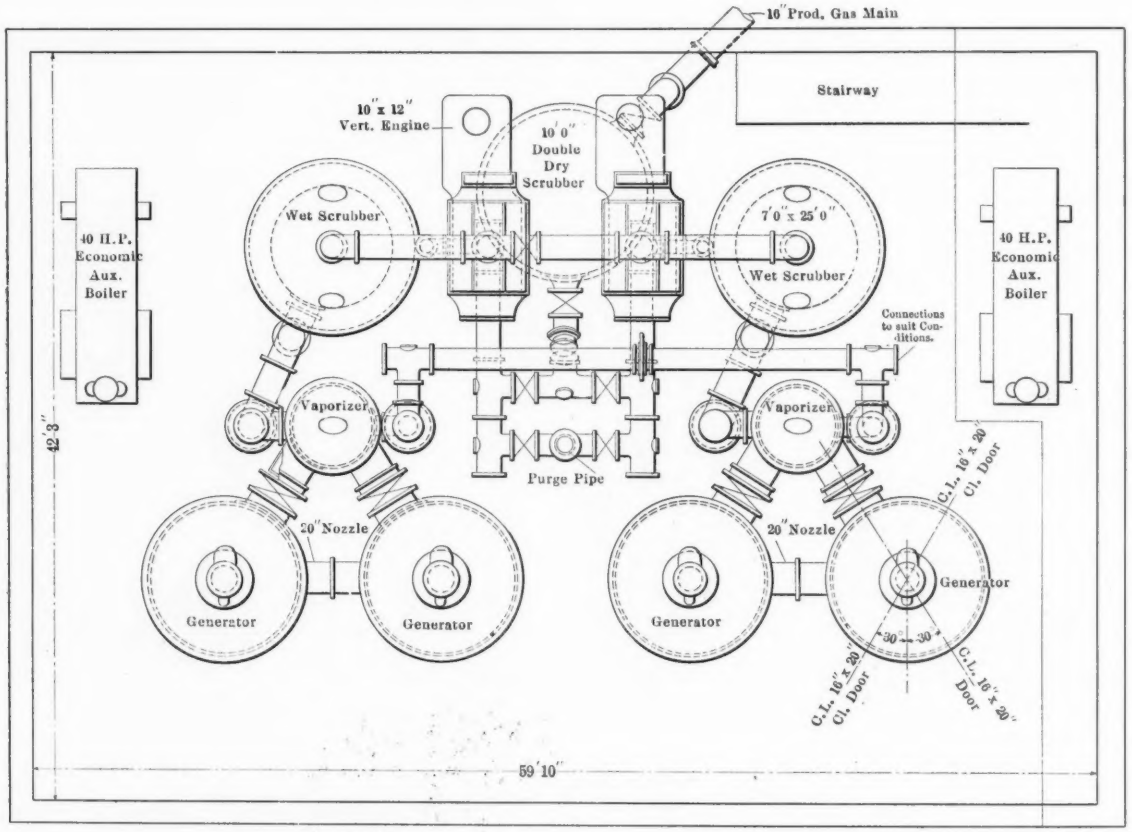


FIG. 2. PLAN OF THE PRODUCER EQUIPMENT

Cooling System and Lubrication

The cooling water is carried from the engine jackets to a "hot well," from which a centrifugal pump takes it for delivery to a system of cooling sprays which deliver it to a surface pool, as shown in Fig. 6. Another pump, of one-fourth the capacity of the spray-feed pump, takes water from the pool to the engine jackets. These two pumps are direct-driven by a single motor and their relative work is therefore constant. The pool overflows into the "hot well," and as the spray pump works four times as hard as the jacket pump, the cooling water is circulated through the sprays four times as often as it passes through the engine jackets.

The cylinders of the engines are sup-

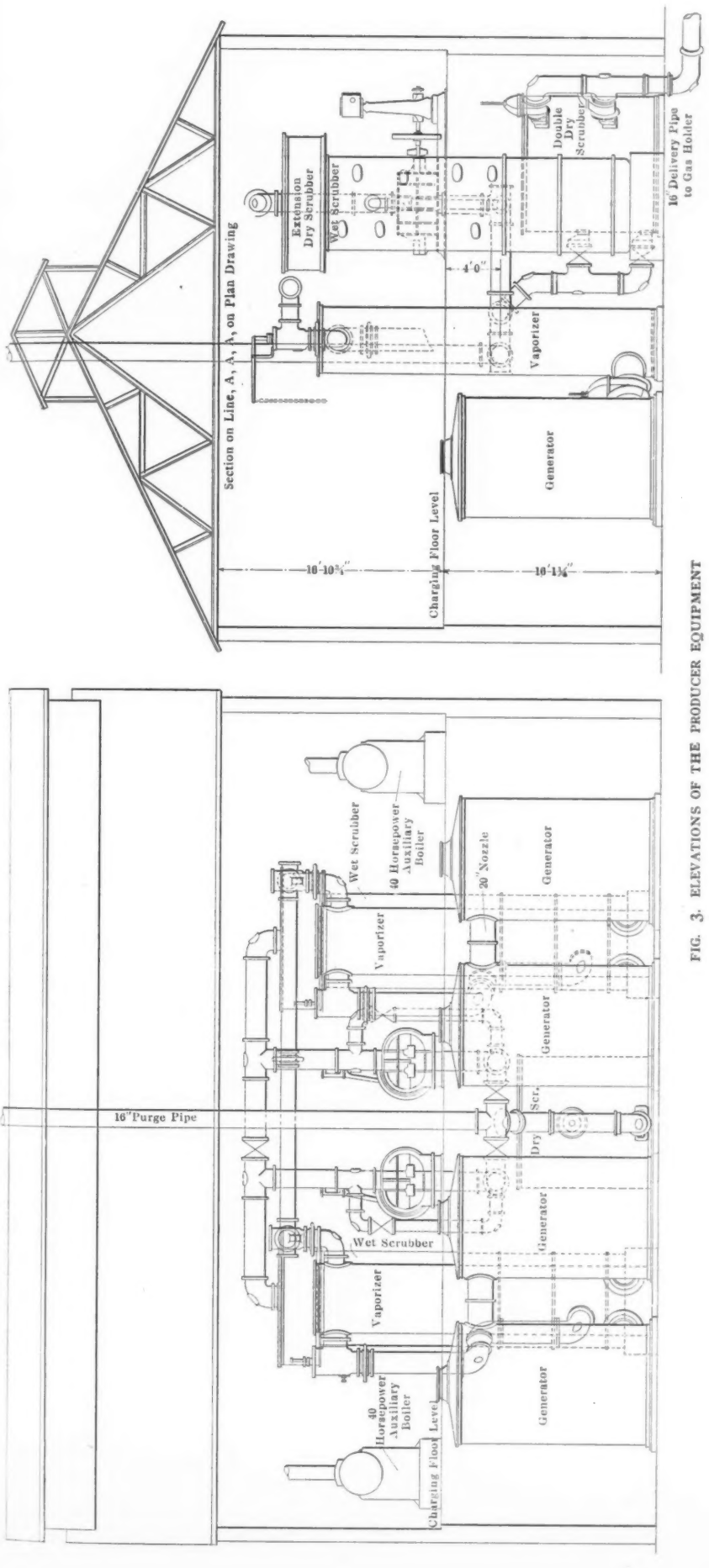


FIG. 3. ELEVATIONS OF THE PRODUCER EQUIPMENT

plied with oil by force-feed lubricators of the familiar Milwaukee make.

Plant Performance

The load from 6 o'clock in the morning until dusk ranges from 400 to 600 kilowatts usually. Of this, about 100 kilowatts consists of stationary motors and a little of it is in incandescent lamps; the bulk of it is the railway load. From dusk until 9 p.m. the railway load is about 600 to 750 kilowatts, the motor load negligible and the lighting load about 250 kilowatts. From 9 o'clock until 1 the load falls off steadily, and from 1 o'clock until 6 the next morning it is about 75 kilowatts. From this it will be evident that the principal load during eighteen or nineteen hours of the twenty-four is of the worst class — a railway load imposed through the medium of rotary converters. There seems to be no difficulty, however, in operating the main units in parallel during the heavy-load periods or in running one of them with a relatively large converter during the other periods. When the 500-kilowatt converter is running with one of the generating units it does not behave in an ideal manner, probably because its flywheel effect is so great in proportion to that of the generating unit. The irregularity is slight, merely causing a slight instability in the power factor of its own circuit, causing no practical difficulties.

RECORD OF TOTAL-OUTPUT WATT-HOUR METER, ETC.

Date.	Total Output, Kw.-Hours.	Total Pounds Coal.	Lb. of Coal Per Kw.-Hour.
Sept. 1	8,300	17,698	2.12
Sept. 2	8,400	16,672	1.86
Sept. 3	8,200	20,890	2.62
Sept. 4	8,200	20,435	2.49
Sept. 5	8,900	19,278	1.95
Sept. 6	7,800	17,350	2.22
Sept. 7	8,100	18,890	2.32
	57,900	131,214	Average, 2.00

One engine is run alone from about 10 or 11 o'clock at night until the "peak" begins to show up the following afternoon; then the other engine is put in commission and the two units operated in parallel until the heavy load subsides, about 10 or 11 o'clock. The engine which ran through the previous day is then shut down and the other one continues until after the heavy-load period of the following day, assisted by the first engine.

The switchboard is not provided with a recording wattmeter, and there is therefore no record of the load fluctuations. The accompanying table gives the record of the total-output watt-hour meter for the first week in September, and the corresponding coal consumption in the producer house, including the loss due to shutting down one producer for cleaning and starting up the other, which occurred during that week. The average is not

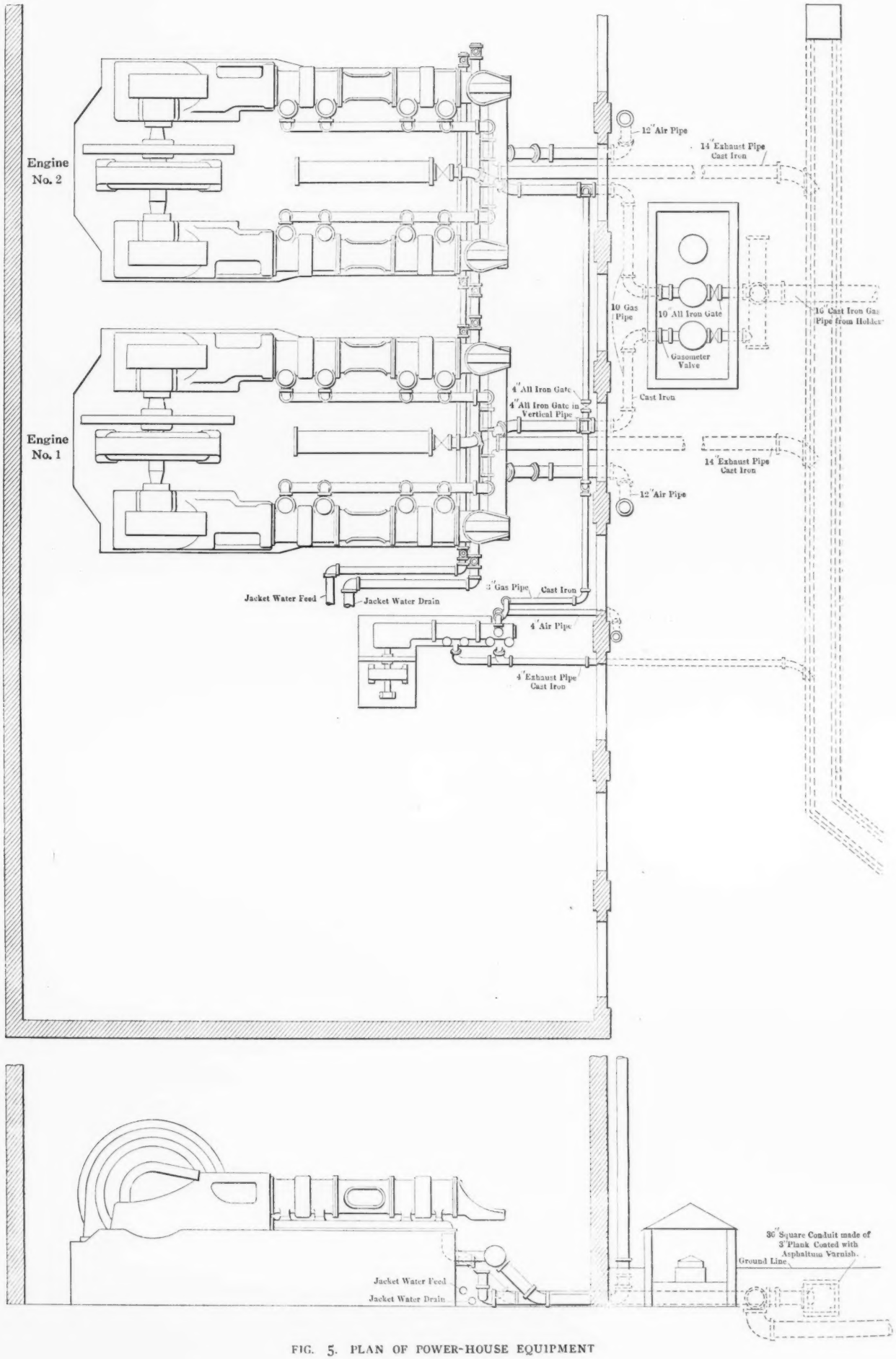


FIG. 5. PLAN OF POWER-HOUSE EQUIPMENT

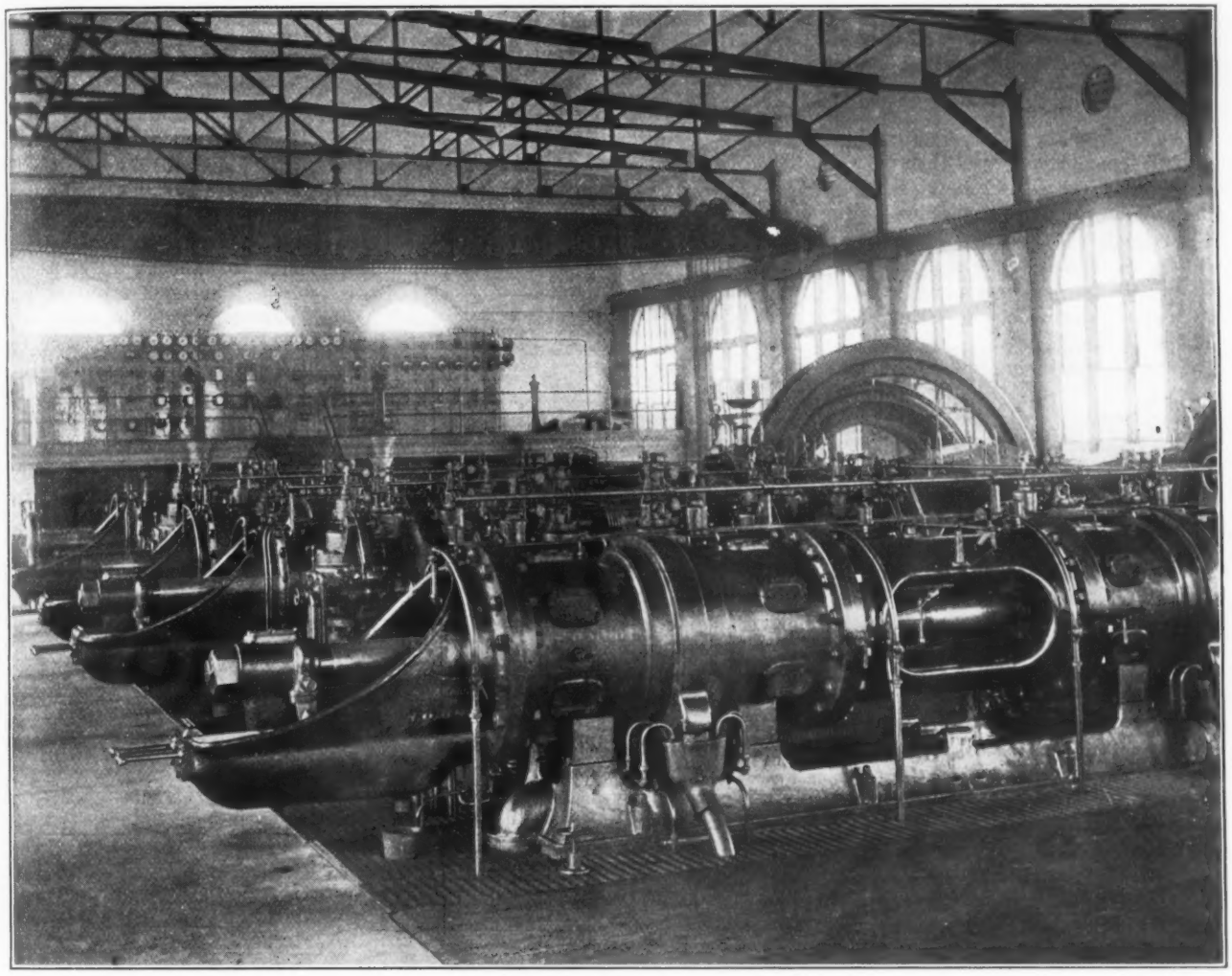


FIG. 4. VIEW OF POWER HOUSE LOOKING TOWARD SWITCHBOARD

extravagant, considering the light load carried every night from midnight on. The plant records for October 15, which was circus day, with a very heavy railway load, show 11,130 kilowatt-hours for the twenty-four hours, and a coal consumption of 16,795 pounds, or 1.51 pounds per kilowatt-hour.

Fig. 7 is an indicator diagram taken while the generator was delivering 748 kilowatts at the switchboard. The data as to the running conditions and the diagram measurements are inscribed on the diagram. From these figures the combined conversion efficiency of the unit was

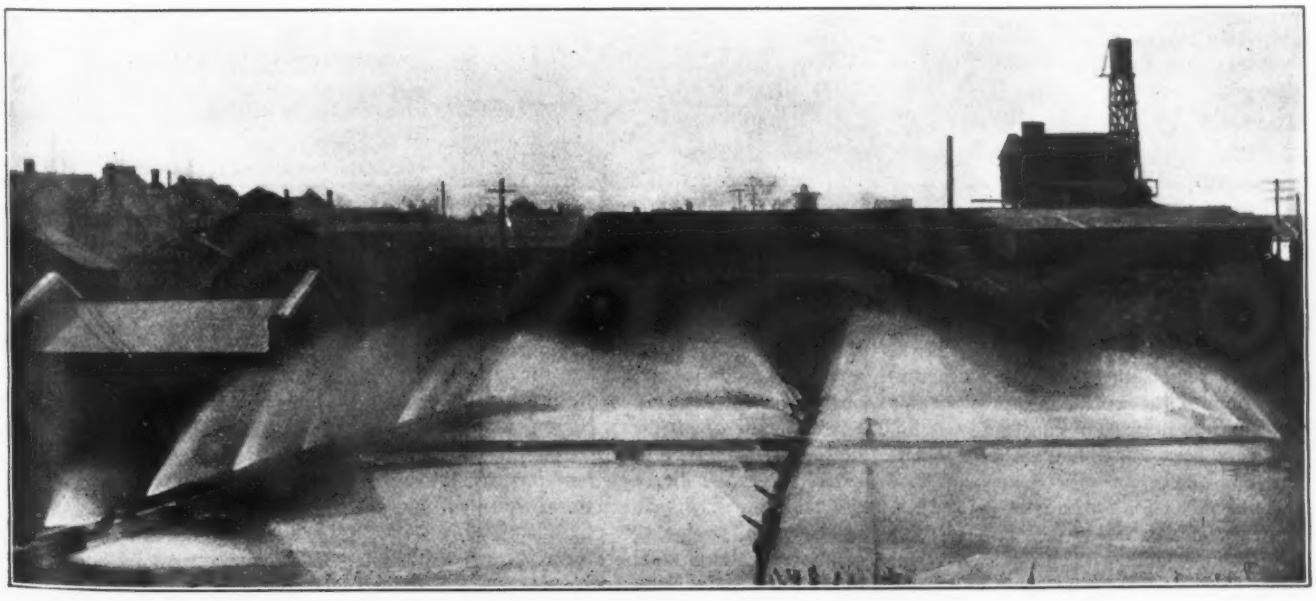


FIG. 6. SPRAYS COOLING THE JACKET WATER

about 80 per cent., if it be assumed that the same power was developed in each end of each one of all four cylinders, which is a somewhat rash assumption. Fig. 8 is a diagram taken at 420 kilowatts of generator load. Both diagrams are unusually smooth and well-shaped, and are reproduced here precisely as they came from the indicators. They appear

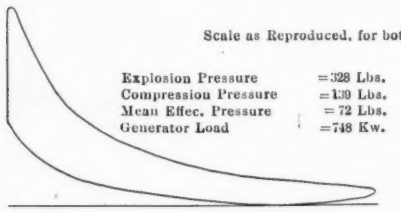

Scale as Reproduced, for both Diagrams = 300 Pounds per Inch
Explosion Pressure = 328 Lbs.
Compression Pressure = 139 Lbs.
Mean Effec. Pressure = 72 Lbs.
Generator Load = 748 Kw.

FIG. 7. FULL-LOAD INDICATOR DIAGRAM

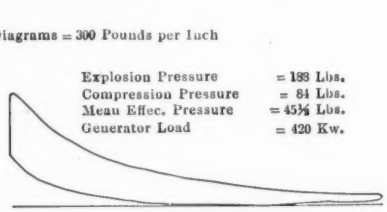


FIG. 8. MODERATE-LOAD INDICATOR DIAGRAM

to indicate that the opening of the exhaust valves occurs a trifle late and that the ignition could have been advanced a little farther. However, the indicator diagram cannot be taken alone as a criterion, and criticising these diagrams would be work for a cynic of the sourest type.

The writer is indebted to E. D. Latta, Jr., of the "Four C's", and H. McIntyre, erecting engineer for the engine builders, for many courtesies and much information tendered during his visit to the plant. The company's records and employees were placed at his disposal as far as desired, and the operating history of the plant was laid bare without reserve by Mr. McIntyre, by permission of the home office. The broken oil pipe, a brief siege of governor trouble at the outset, and an occasional back-fire were the only skeletons in the closet, however.

At the eleventh annual convention of the American Mining Congress, held at Pittsburg, Penn., recently, Dr. I. C. White, State geologist of West Virginia, said, among other things: "It is quite certain that Pennsylvania will not furnish more than forty billion tons and Ohio probably not more than twenty-five billion tons of commercial bituminous coal. This shortage in coal brings to the citizens of the Pittsburg region, the present manufacturing center of the world, the most serious problem that has ever confronted them. You have been told that you had 430 billion tons of coal in your mines and that it would suffice for 150 to 200 years, while the truth is you have only one-third of that amount, and with present wasteful mining methods it will last only fifty years."

The Current and Output of Alternating Current Motors

By Paul C. Percy

The relation between the current taken by a single-phase alternating-current motor and the output of the machine is exactly the same as the relation between current and output of a direct-current motor of the same voltage and efficiency. Thus, the output in brake horsepower of a direct-current motor is equal to the current multiplied by the voltage, multiplied again by the efficiency and divided by 746. The brake horsepower of a single-phase motor is equal to the product of volts, amperes, efficiency and power factor, divided by 746. Representing these relations by a formula brings out the similarity of the two cases more distinctly. Let

E = Electromotive force of the circuit,
I = Current taken by the motor,
e = Conversion efficiency of the motor,
p = Power factor of the motor.

Then

$$\frac{E \times I \times e \times p}{746} = \textit{Brake horsepower},$$

and this formula applies to the direct-current motor as well as the other. The power factor of all direct-current motors, however, is 1; therefore it is never considered, and the direct-current formula is the one above with the letter p omitted.

The case of the two-phase motor is somewhat different. There are four wires in a two-phase motor circuit, and the circuit is really two single-phase circuits supplied from a single generator. The two-phase motor is provided with two distinct windings, one taking current from one pair of the line wires and the other from the other. The "current" of any motor is always the current in any *one* of the wires leading to the motor. Thus, if a two-phase motor takes 25 amperes from each division of the supply circuit, the motor "current" is said to be 25 amperes, not 50, although the effect of the two 25-ampere currents upon the power of the motor is exactly the same as that of 50 amperes in a single-phase motor.

The two-phase case is illustrated diagrammatically by Fig. 1, where the generator windings are indicated by the letters A, A and B, B, and the motor windings by a, a and b, b. The two circuits A' and B' constitute a two-phase circuit, but they are actually two simple circuits supplied from two simple windings on the generator armature. The total power delivered to the motor is twice the power transmitted by each of the two circuits, and the power in each of these is computed exactly as for a single-phase circuit. The "voltage" of the motor is that measured across either pair of the four leads connecting it with the circuit.

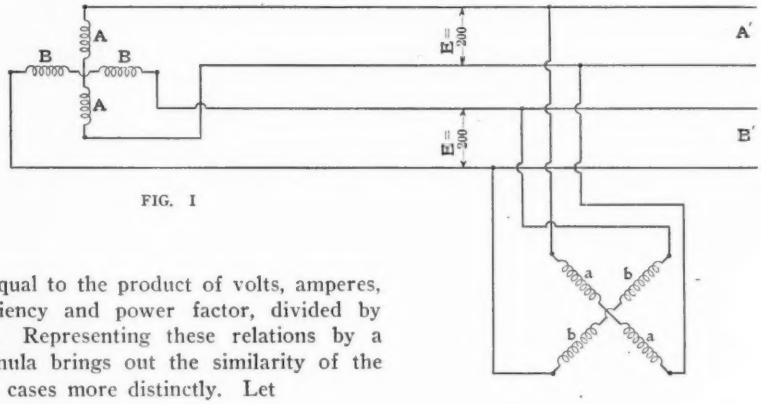


FIG. 1

From the foregoing it is probably evident that the formula for the output of a single-phase motor can be applied to a two-phase motor by multiplying the numerator by 2, thus:

$$\frac{2 \times E \times I \times p \times e}{746} = \textit{Brake horsepower}.$$

There is no need, however, for a coefficient in both the numerator and denominator of the fraction. It is better to reduce it to either

$$\frac{E \times I \times p \times e}{373} = \textit{Brake horsepower},$$

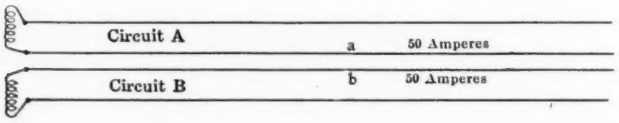


FIG. 2

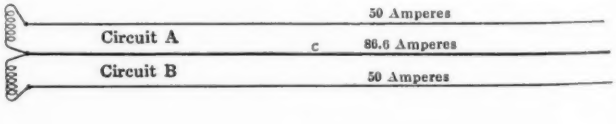


FIG. 3

or else

$$0.00268 \times E \times I \times p \times e = \textit{Brake horsepower.}$$

The case of the three-phase motor is considerably different, on account of the peculiar characteristics of the three-phase circuit. This comprises three wires, but the circuit does not consist of three independent single-phase circuits supplied from a single generator; consequently, the current in each wire is not one-third of the current that a single-phase motor of the same intake would require. The three-phase circuit is the equivalent of three single-phase circuits with one wire of each merged into one wire of each of the other two circuits. Consequently, each wire carries more than the equivalent single-phase current—that is, the current it would carry if there were three independent circuits. The exact relation may be readily understood from the following analysis:

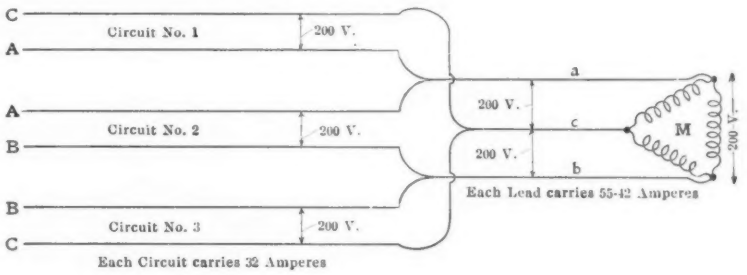


FIG. 4.

If two wires of two separate circuits carrying equal alternating currents which are not in phase or "step" with each other be combined into one wire, the resultant current in that wire will not be equal to the arithmetical sum of the two original currents, but will be equal to that sum multiplied by the cosine of half the "angle" of phase difference existing between the two original currents. For example, if the two separate wires carry equal currents, and the two currents are out of phase with each other one-third of a cycle, or 120 degrees, the current in the wire formed by combining the two original wires will be

$$I = (i + i) \times \textit{cos.} \left(\frac{120}{2}\right),$$

or

$$I = 2 \times i \times \textit{cos.}\ 60.$$

The cosine of 60 degrees is 0.866, and twice this is 1.732; consequently the formula may be condensed into

$$I = 1.732 \times i,$$

I being the current in the consolidated wire and i that in each of the original wires. It must be remembered that this formula applies only to two equal currents 120 degrees out of phase with each other. Figs. 2 and 3 illustrate this case. In Fig. 2 each of the circuits carries 50 amperes; consequently the current in the wire a is 50 amperes, and that in the wire b is equal. Now if these two wires be displaced by a single wire, as in Fig. 3, the current in that wire will be

$$I = 1.732 \times 50 = 86.6$$

amperes.

Each of the wires of a three-wire three-phase circuit, as already explained, is the equivalent of the consolidation of two wires of two single-phase circuits, of which there are three, and the current in each wire of the actual circuit is the equivalent of a consolidation of the currents in two of the imaginary single-phase circuits which are 120 degrees out of phase with each other. Consequently, the current per wire in the three-phase circuit is 1.732 times the current per circuit in three single-phase circuits having the same phase relations and carrying the same total power.

Now the watts in three single-phase circuits are equal to $3 \times E \times I \times p$, and since the current per wire in a corresponding three-phase circuit is 1.732 times that in each of the single-phase circuits, the watts in the three-phase circuit are equal to

$$\frac{3 \times E \times I \times p}{1.732}.$$

But $3 \div 1.732 = 1.732$; consequently, the watts in a three-phase circuit are equal to

$$1.732 \times E \times I \times p,$$

and the output of a three-phase motor is

$$\frac{1.732 \times E \times I \times p \times e}{746} =$$

Brake horsepower.

This reduces to the somewhat more convenient expression:

$$\frac{E \times I \times p \times e}{430.7} = \textit{Brake horsepower.}$$

Perhaps a numerical example will make the foregoing explanation a little clearer. Suppose there were three single-phase circuits supplied from a single generator in such a way that the currents in the circuits were 120 degrees out of phase with each other. Further suppose that a three-phase motor were connected to these three circuits by three leads connected, as shown in Fig. 4, where the motor windings are represented by the three interconnected helices at M, the three single-phase circuits are indicated by straight lines and numbered 1, 2 and 3, and the three motor leads are identified by the letters a, b and c. The motor leads are plainly nothing more or less than consolidated extensions of the main circuit wires; the currents in the wires A and A combine in the lead a; those in the wires B and B combine in the lead b, and those in C and C combine in c.

If the voltage of each of the single-phase circuits is 200, the voltage between each pair of motor leads will be 200, as the figures on the diagram indicate. If the load on the motor is such as to cause 32 amperes to flow in each supply circuit and the power factor of the motor is 90 per cent., the total power supplied by all three circuits will be

$$3 \times 32 \times 200 \times 0.90 = 17,280$$

watts, and if the efficiency of the motor is 90 per cent., the output will be

$$\frac{17,280 \times 0.90}{746} = 20.85$$

brake horsepower.

Now, according to the explanation previously given, if each supply circuit carries 32 amperes, each of the motor leads will carry $1.732 \times 32 = 55.42$ amperes. The output, therefore, according to the three-phase formula, will be

$$\frac{200 \times 55.42 \times 0.90 \times 0.90}{430.7} = 20\ 85$$

brake horsepower.

Since the three single-phase circuits are interlinked by the motor leads a, b, c, they may as well be consolidated throughout by substituting one large wire for the wires A and A, another for B and B, and a third one for C and C. Fig. 5 shows the result, which is an ordinary three-phase circuit.

If the reader unfamiliar with alternating-current phenomena will keep it steadfastly in mind that the current (I) is always the amperes per wire, regardless of the type of motor or circuit, he will avoid much confusion and uncertainty. When a beginner tries to add together alternating-current values he immediately gets into trouble.

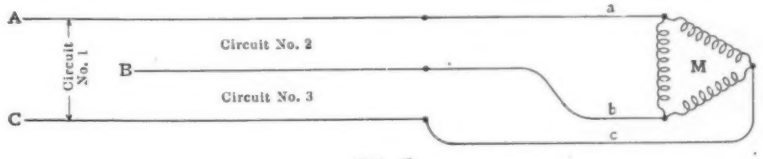


FIG. 5

Unnecessary Losses in Firing Fuel Oil*

Valuable Data Relating to Fuel Oil, with a Plea for an Automatic System of Control to Offset Losses Due to Excess Air Supply

BY C. R. WEYMOUTH

Practically all oil-fired boiler plants in stationary practice are subject to hand control throughout. It is customary to maintain a uniform oil pressure at the oil pump and in the oil-pressure main, and to throttle the supply of oil by hand at all of the individual burners. It is also customary to operate with full boiler steam pressure on the main supplying steam to all the burners, and to regulate by hand the supply of steam for atomizing purposes, at each of the individual burners. Boiler dampers also are all subject to hand control on the individual boilers.

2. In a central station having, say, twenty 500-horsepower boilers there would would be about 60 burners. For economy of labor, there would probably be not more than two or three firemen to the shift, in a plant of this size. On a commercial railway or lighting load subject to the usual fluctuations, such a plant would probably be operated with the rear boiler dampers clamped in fixed positions, wide open or nearly so. The supply of steam to burners would receive little attention, but the supply of oil to the burners would be regulated for variations in load by throttling to the extent necessary for maintaining the desired steam pressure. In such a plant there would be a more or less uneven rate of firing at the various boilers, and an excess of air for combustion at all loads, particularly at the lighter ones corresponding to a nearly uniform rate of flow of air through the furnace. The operators are likely to become careless, not noticing the drop in steam pressure with a sudden increase in load until this has become considerable, necessitating a severe momentary rate of firing in a number of boilers to bring the steam pressure back to the normal. This severe duty increases the expense for repairs to the boiler settings, rate of burning out of tubes, etc.

3. In certain plants where engineers are enlightened as to the principles of combustion, the attempt is frequently made to operate on a reduced air supply, with the result, if the dampers are set for mean or nominal load, that the chimneys smoke excessively on overloads before the limited number of firemen can reach all the dampers to open them.

4. As the lamentable result of these conditions, the average boiler-plant efficiency with crude oil, even with the best types of boiler, averages much nearer 70 than 80 per cent., which is possible in large plants under proper methods of control.

5. Probably it will always be impossible to instill into the mind of an ordinary fireman such knowledge of the principles of combustion and the losses due to excess air supply, as to obtain economical results in large stations where it is necessary to depend on hand firing. Improved conditions can be secured by the employment of a boiler-room engineer whose duty it is to scrutinize all fires from time to time and to coach the firemen in their duties; but the only ideal method seems to be an automatic system of control, where the various adjustments having once been made for economical conditions are automatically repeated for the various conditions of load, maintaining a high average economy from month to month. With well designed oil furnaces and careful adjustment under uniform load conditions, carefully conducted tests have shown that it is possible to obtain high percentages of CO_2, indicating as low as 10 per cent. excess air over the requirements for perfect combustion, with no unconsumed elements in the flue gases.

6. Numerous data relating to coal fuel are available, showing the importance of reduced air supply as tending to high furnace efficiency; also the relation of excess air supply to any observed percentage of CO_2 and other factors of gas analysis. As few data for oil fuel are available, the following will be presented:

7. All Pacific coast crude oils contain a certain amount of moisture, sulphur, nitrogen and oxygen; the main constituents being carbon and hydrogen. The characteristic difference in oils of different gravities lies in the relative quantities of carbon and hydrogen contained, there being more carbon and less hydrogen in the heavier oils, less carbon and more hydrogen in the lighter. In the better grades of oil treated at the wells before shipment, in which moisture has been largely eliminated, it may be roughly assumed that 3 per cent. of the oil is made up of sulphur, nitrogen, oxygen and water. This relationship is not universal, certain southern California oils containing a large percentage of sulphur.

8. The predominating oil used on the Pacific coast, known as Bakersfield oil, averages about 16 degrees Baumé, which is equivalent to 336 pounds of oil per 42-gallon barrel. The ultimate analysis of this oil is about as follows: Carbon, 85 per cent.; hydrogen, 12 per cent.; sulphur, 0.8 per cent.; nitrogen, 0.2 per cent.; oxygen, 1 per cent.; water, 1 per cent. A number of lighter oils in general use, ranging in the neighborhood of 18 to 20 degrees Baumé, would average about as follows: Carbon, 84 per cent.; hydrogen, 13 per cent.; sulphur, 0.8 per cent.; nitrogen, 0.2 per cent.; oxygen, 1 per cent.; water, 1 per cent. Certain heavier oils ranging from 12 to 14 degrees Baumé average about as follows: Carbon, 86 per cent.; hydrogen, 11 per cent.; sulphur, 0.8 per cent.; nitrogen, 0.2 per cent.; oxygen, 1 per cent.; water, 1 per cent.

TABLE I. WEIGHT OF AIR REQUIRED FOR COMBUSTION OF OIL OF DIFFERENT GRADES.

Grade of Oil.	Light.	Medium.	Heavy.
Per cent. of C	84.00	85.00	86.00
Per cent. of H	13.00	12.00	11.00
Per cent. of S	0.80	0.80	0.80
Per cent. of N	0.20	0.20	0.20
Per cent. of O	1.00	1.00	1.00
Per cent. of H_2O	1.00	1.00	1.00
Calculated air per pound of oil chemically required—pounds	14.25	14.02	13.79
Corresponding maximum per cent. CO_2 by volume—dry gases of combustion, per cent	15.16	15.52	15.89

9. As a result of tests by Edmond O'Neill, professor of chemistry of the University of California, the calorific value of Bakersfield oil may be taken as about 18,600 B.t.u. per pound, allowing for the presence of about 1 per cent. moisture as indicated above. When corrected for moisture the net heat units per pound of oil are proportionally higher, although there is a slight loss in furnace efficiency due to the presence of moisture, inasmuch as all such water is evaporated into steam and superheated to the temperature of the escaping gases, involving an amount of heat both sensible and latent.

10. On the basis of the above analysis,

*Presented at the annual meeting (December, 1908) of the American Society of Mechanical Engineers.

the chemical requirements of air for complete combustion per pound of oil are as shown in Table 1.

11. In the various textbooks, the values given range from 16 to 18 pounds of air per pound of oil, but an average of 14 pounds of air per pound of oil is more nearly correct.

12. The ordinary method of indicating and measuring steam to atomize oil has been to express the quantity as a percentage of the actual amount of water evaporated in the boiler. This percentage ranges from about 2 to 5 and over, depending on the system of oil burning, type of burner, etc. While such a percentage rating is no doubt convenient, it is inaccurate, in that the steam consumption of oil burners is proportional to the oil burned and not to the water evaporated. Various tests have shown that the steam consumption of oil burners ranges from 0.14 to over 0.5 pound of steam per pound of oil. The average value of good performance is about 0.3 pound of steam per pound of oil, although with hand regulation on variable load this quantity should be slightly increased, and is somewhat dependent on the gravity of the oil, temperature at the burners, etc. In stationary practice, the use of air for atomizing purposes has been practically abandoned.

13. As no direct experiments have been made showing the loss in boiler efficiency due to various percentages of excess air supply, the writer will present some simple calculations showing the amount of this loss.

14. It is well known that the loss due to an excess of air supply is not only on account of the direct loss in heating the added air to the temperature of the flue gases, but there is a secondary loss due to the fact that corresponding to an excess of air, there results a higher flue temperature not only for the actual amount of air necessary for combustion, but for all such excess air. Calculations as to boiler performance are simplified with oil fuel, as practically complete combustion is secured in a well designed furnace, the carbon and carbon monoxide usually being burned to CO_2. The stack losses include the sensible heat contained in the dry gases of combustion, the sensible and latent heat in the steam from the combustion of hydrogen and oxygen and in the steam introduced through the burner, and the moisture present in air for combustion.

15. Assuming complete combustion, and employing a boiler radiation loss of 3 per cent., the writer has calculated the boiler efficiency, at rating, for various percentages of excess air supply, as given in Table 2.

The 3 per cent. used for boiler radiation is subject to some variation, being greater in small boilers and less in large units. For medium units, 3 per cent. is probably very close.

TABLE 2. BOILER EFFICIENCY FOR EXCESS AIR SUPPLY.

Excess Air Supply, Per Cent.	10	50	75	100	150	200
Assumed temperature escaping gases, deg. F....	400	450	475	490	Over 500	Over 500
Corresponding ideal efficiency of boiler, per cent.	84.2	80.27	77.66	75.22	Under 70.94	Under 67.09
Possible saving in fuel due to reduction of air supply to 10 per cent. excess, expressed as per cent. of oil actually burned under assumed conditions.....	0	4.67	7.78	10.68	Over 15.76	Over 20.32

16. The stack temperatures for any particular type of boiler, for any given load and corresponding to any assumed per cent. of excess air, will vary with the size of boiler, arrangement of heating surface, character of baffling, condition of heating surface, etc. Stack temperatures will also vary with the different types of boiler corresponding to these factors. The temperatures given corresponding to the stated air supply, from 10 to 100 per cent. excess, are those to be expected in ordinary practice and necessarily approximate: with boilers having three passes of gases and sinuous headers, the temperatures in general will be lower than those indicated; with boilers having but one pass and flow of gases parallel to tubes, the temperatures in general will be higher than indicated.

17. Very few data are available for the temperatures corresponding to 150 and 200 per cent. excess air, and the corresponding figures are given merely to show in a general way the magnitude of the losses easily resulting from careless firing of crude oil. The flue temperatures assumed are also subject to variation dependent on the rate of forcing the boiler and other well known elementary factors. The excess air with careless oil burning is usually greater than with careless coal firing, because of the greater excess draft power of chimneys. In Table 2 the writer has calculated the saving that could be effected by reducing the air supply from that specified to an ideal condition assumed to correspond to 10 per cent. excess air. This saving in fuel is of vast importance, but has been almost completely neglected with oil fuel.

18. It is possible to obtain a fair notion of the percentage of excess air from a mere determination of the amount of CO_2; that is, assuming all hydrogen having been burned to H_2O and all carbon to CO_2. Any simple formula involving the element CO_2 must be dependent on an assumed percentage of hydrogen in the oil fuel, but inasmuch as the hydrogen contained is fairly uniform for any given grade of oil, there is but little error in such an assumption.

19. Table 3 shows the calculated weight of air per pound of oil and the ratio of actual air supply to chemical requirements, for the various grades of oil and various percentages of CO_2. Under the present systems of firing the amount of CO_2 present in the flue gases is often as low as 4 or 5 per cent. With an ample supply of labor and a careful and scientific adjustment of dampers by hand, the percentage of CO_2 under an ideal and uniform load can be maintained as high as 13 per cent. With automatic control and under variable load conditions it has been found possible to maintain a high percentage of CO_2 conforming very closely to ideal conditions.

TABLE 3. POUNDS OF AIR PER POUND OF OIL AND RATIO OF AIR SUPPLIED TO THAT CHEMICALLY REQUIRED.

Per Cent. CO_2 by Volume as Shown by Analysis Dry Chimney Gases.	Light Oil. C, 84 per cent.; H, 13; S, 0.8; N, 0.2; O, 1; H_2O, 1.		Medium Oil. C, 85 per cent.; H, 12; S, 0.8; N, 0.2; O, 1; H_2O, 1.		Heavy Oil. C, 86 per cent.; H, 11; S, 0.8; N, 0.2; O, 1; H_2O, 1.	
	Pounds of Air Per Pound Oil.	Ratio Air Supply to Chemical Requirements.	Pounds of Air Per Pound Oil.	Ratio Air Supply to Chemical Requirements.	Pounds of Air Per Pound Oil.	Ratio Air Supply to Chemical Requirements.
4	51.04	3.607	51.93	3.704	52.45	3.803
5	41.31	2.899	41.71	2.975	42.12	3.054
6	34.58	2.427	34.90	2.490	35.23	2.554
7	29.77	2.089	30.04	2.143	30.31	2.198
8	26.17	1.836	26.39	1.883	26.62	1.930
9	23.37	1.640	23.56	1.680	23.75	1.722
10	21.12	1.482	21.29	1.518	21.45	1.555
11	19.83	1.391	19.43	1.386	19.58	1.419
12	17.76	1.246	17.88	1.276	18.01	1.306
13	16.46	1.155	16.57	1.182	16.69	1.210
14	15.36	1.078	15.45	1.102	15.55	1.127
15	14.39	1.010	14.48	1.033	14.57	1.056

The rest of the paper was mainly devoted to a description of an automatic fuel-oil regulating system designed by the author and patented by Charles C. Moore & Co., of San Francisco. It will be described in the "new-appliance" department of this paper next week.

Governing Gas Engines

By L. B. Lent

In talking to a large number of men, I find that there is considerable misconception, or possibly lack of information, regarding the limitations of the governor on a gas engine. A great many who are interested have a rather vague idea that a governor is that part of an engine which controls the speed, and that if the speed is not uniform, the fault is necessarily that of the governor. In many cases this is not true, and in gas-engine practice it is more often not true than otherwise.

A large percentage of gas engines are installed to drive electric generators, and if the regulation of the engines is good enough to permit engine-type alternators of moderately high frequency to be operated in parallel, all the requirements of close regulation are supposed to have been met.

The question of the "system" of governing is relatively unimportant in this discussion. This subject has been treated very thoroughly in Power and The Engineer, in several excellent articles by Mr. Junge and others. It is therefore assumed that the reader knows at least the general facts regarding the so-called "quality," "quantity" and other systems of speed control.

The governor on any gas engine is supposed to measure the energy of the charge delivered to the cylinder or cylinders, but, with possibly one exception, it has no control of the charge after it is in the cylinder. The exception is that in which the governor controls the time of ignition. The form of the indicator diagram is considerably influenced by varying the ignition point, and by this means the mean effective pressure may be affected by the governor after the charge is in the cylinder. In most engines, however, the governor does not control the ignition and in this discussion it is assumed that it does not.

If the mean effective pressure is proportional to the load, the speed should remain constant. The duty of the governor is to control the mean effective pressure, which means primarily the control of the energy developed by the combustion of the charge admitted. Whether the governor measures the "quality" or "quantity" of the mixture, it actually measures volumes or pressures of either the gas or air or both. And whether it throttles the charge, cuts off the charge, or throttles or cuts off the gas alone, a variation of gas pressure in the pipe leading to the engine exerts an influence which the governor cannot control completely.

It should be easily understood that if the gas pressure should increase or decrease the proportions of gas and air will be materially affected. The amount of gas which will flow through an opening of given size in a certain length of time depends on the pressure causing the flow. Air enters the engine intake almost always at a pressure slightly less than atmospheric and at a very constant pressure. The effect of variations in the gas pressure is obviously variation of the heat value of the charge, and this affects the mean effective pressure regardless of the governor. This variation in gas pressure is more often met in producer plants and should be carefully guarded against. So also should variations in gas quality, which is likewise common to producers. The effect of such variation is similar to that of pressure variations.

Here, then, are conditions which are not under the governor control and which exercise considerable influence on the mean effective pressure developed, and therefore, even though all conditions are correct and the governor operation is faultless, the charge delivered will not produce the energy for which the governor has measured out the charge or its proportions.

It is well known that the charge after delivery does not always burn as it should. It must be quite obvious that such occurrences as back-fires, misfires and pre-ignitions cannot be called normal or proper, and it must also be quite apparent that such things discount the proper action of any governor.

In the operation of alternators in parallel, the angular variation per revolution must not exceed a certain amount (about five electrical degrees), or the generators are liable to go out of step. A uniform angular velocity is therefore highly desirable, and an approximation to this is secured by a proper relation of crank effort and flywheel effect. If the revolutions per minute remain constant, the variable factor in the development of the crank effort is the mean effective pressure and this is the factor over which the governor exercises control. Its control ceases, however, after the charge is delivered to the cylinder, and does not extend to the point of cutoff as in a Corliss steam engine. Hence the governor's influence on the form of the crank-shaft curve depends largely on the number of times it is allowed to exercise that control; in other words, on the number of power impulses per revolution.

Since power is developed and delivered to the engine shaft intermittently, no matter how many times per revolution, it is essential that the power be delivered regularly, particularly if there are but few impulses per revolution. Now if faulty combustion prevents a regular delivery of power, the resultant effect is no fault of the governor. A decrease in speed due to a misfire, for instance, will cause the governor to seek a new position, which should result in a more powerful charge in the succeeding stroke. But this is an erroneous demand and warranted not by any change of load, but of speed, and more often causes hunting than otherwise. As a matter of fact, the flywheels of gas engines are built so heavy that one missed explosion does not materially decrease the speed, but only a very slight change in speed is necessary to throw out of step a pair of alternators.

Back-fires are even worse than misfires, because gas coming to the engine may be burned and the ill effect extended over more than one stroke.

Even worse than either of the other influences are pre-ignitions, for in this case power is developed in a negative direction and so produces the most harmful effect on the speed.

The phenomena mentioned are rarely caused by any governor action, and under certain conditions are very common. The causes are now pretty well known and can in a measure be guarded against.

Back-firing, misfiring and even pre-ignition do happen, sometimes frequently, and no governor can prevent their happening nor wholly compensate for it.

The important factor, because it is so variable in the development of the proper crank effort is the mean effective pressure, and if the amount of this can be so extremely varied by accidental agencies, it must be quite clear that the governor is not responsible for resulting speed irregularities.

It may be safely said that any governor which will, under correct conditions, hold a gas engine to a fixed number of revolutions per minute is doing about all that can be expected. And there are very few governors which will not do it when the engine is operating properly.

The conditions of each installation usually present a distinct set of problems, but I believe the reader will readily appreciate that several essential things must be borne in mind in all cases; among them the following are perhaps the more important.

The delivery system of the gas and air should be so arranged that both come to the engine under as nearly unvarying pressures as possible. This applies particularly to the gas pressure, since the air is usually drawn in at a constant pressure, slightly less than atmospheric. If producer gas is used, the quality should be maintained as nearly uniform as possible.

Back-firing, pre-ignition and other faults of engines are usually the results of incorrect design or operation. A full discussion of the causes and prevention of these faults would be too voluminous to present here, but they can be prevented and should be. Dirty gas is a common cause, though not always recognized as such. An extremely weak mixture and the failure of the ignition system are, of course, the most common causes of misfires.

The type of governor in most general use is probably the centrifugal, although the inertia type has been and is being used to a smaller extent. Extreme sensitiveness is not called for, and such governors as the Jahns, Hartung and others of that general class are admirably adapted to the service demanded in gas-engine work.

Indicator Reducing Motion

By Charles W. Barnaby

It is to be hoped that no misguided engineer will construct an indicator-reducing motion on the lines illustrated and described on page 793 of the November 10 issue, for if he does, he will certainly have an unpleasant experience, as the diagrams which will be obtained will be so distorted as to be decidedly misleading and useless.

Placing the pivot point *B*, Fig. 1, of the levers so far above the contact point *E* gives a lever motion which departs very materially from a reduced reproduction of the piston motion of the engine. Suppose *E* to be the position of the contact point of the finger when the piston is at mid-stroke; the finger point would drop to *C* at one end of the stroke, and rise to *D* at the other end. The angular movement of the lever from *E* to *C* is obviously very much greater than from *E* to *D*. This movement being reproduced at the top of the lever which communicates the motion to the indicator drum, a diagram is produced which is shortened at one end and lengthened at the other. That part of the diagram representing the half stroke of the engine at one end is shortened to *e d*, and that part representing the half stroke at the other end is lengthened out to *e c*.

As an illustration of the effect such a rig would have on the diagram which it produced, suppose that an engineer operating a single-valve engine, on which the valve was properly adjusted so as to give a diagram similar to Fig. 2 when a proper reducing motion was used, should apply a rig to his engine made as shown in the illustration referred to. The result would be that although the engine was working under proper steam distribution and should show a diagram with the load properly divded between the two ends, as in Fig. 2, the diagram which he would actually get would be like Fig. 3. He would therefore be justified in coming to the conclusion that his load was not properly divided between the two ends, as cylinder was carrying considerably more than its share of the load. He would then, naturally, shorten or lengthen his eccentric rod, as the case might be, until he had got the points of cutoff apparently somewhere nearly equalized; finally obtaining a diagram similar to Fig. 4.

He now discovers that although the points of cutoff are approximately equal, there is something wrong with his terminal pressures, also with his compression lines. There is obviously something wrong with the clearance—much more clearance space on one end than on the other; but an investigation, involving considerable time and hard work, shows that the clearance is properly divided between the two ends. The facts of the case are that his valve is set away out of center, and, although his faulty rig gives him the diagram, Fig. 4, the valve adjustment which he now has should give him the diagram, Fig. 5, if a proper reducing motion was used.

He now tries a friction diagram. The figure which he gets would not look well in print, ditto his remarks and meditations. Suffice it to say that the resulting diagram is decidedly negative on one end and decidedly positive on the other. An exaggerated case of "freak."

It is obvious that when a sliding contact of the finger is used, the pivot *B*, Fig. 1, should be dropped to the point *F* on the same level as the point of contact *E* of the finger in about mid-position. In case a roller contact is used at *E*, the center of the pivot *F* should, of course, be on line with the center of the roller when in about mid-position. In either case, if the vertical movement *C D* of the point *E* does not exceed one-half to one-third of the length of the lever *F E*, the movement imparted to the indicator drum will be sufficiently correct for ordinary purposes.

In case it is desirable or necessary to keep the pivot *B* above the level of the point of finger contact, the finger movement should be carried up vertically and applied to its lever at a point about on a level with the pivot *B*, in some such manner as shown in Fig. 6.

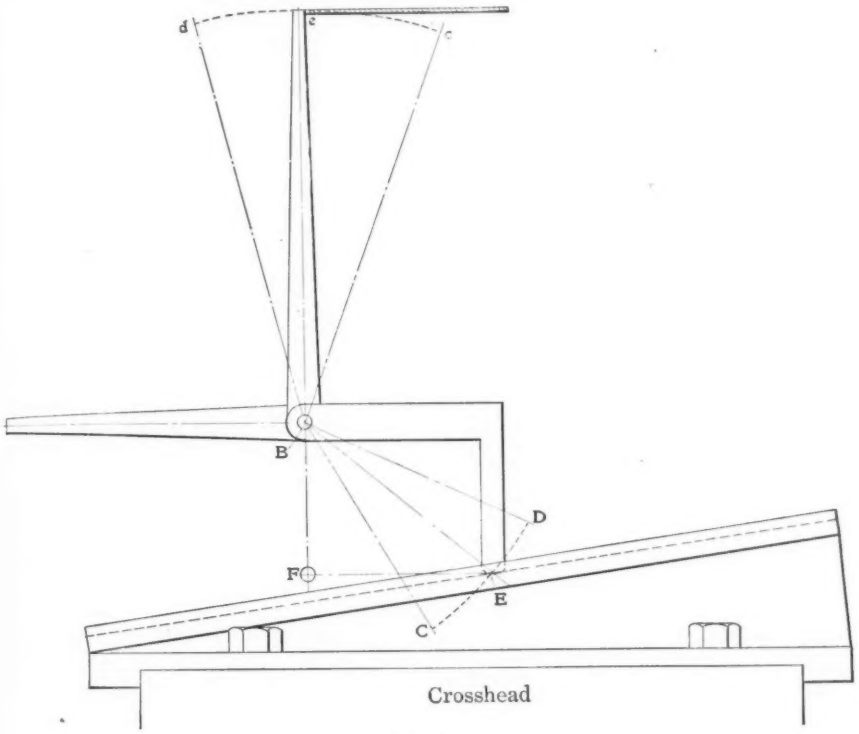


FIG. 1

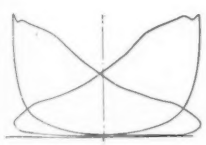


FIG. 2

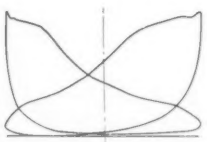


FIG. 3

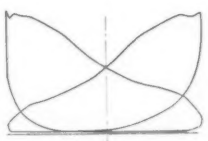


FIG. 4

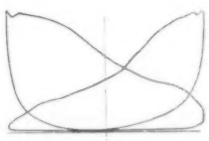


FIG. 5

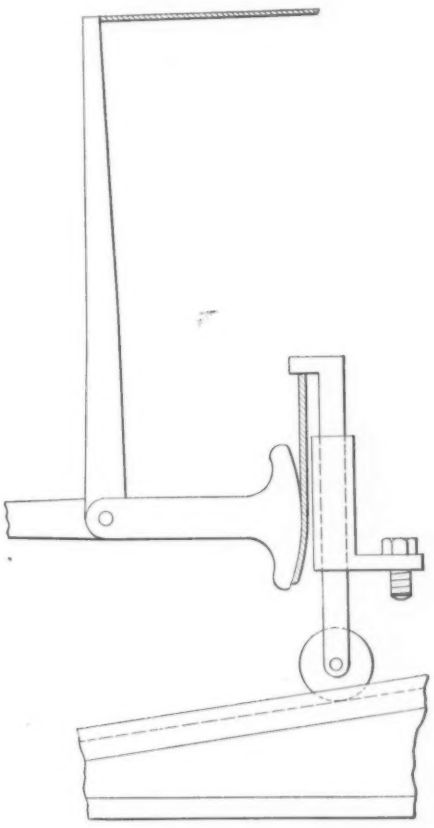


FIG. 6

By Prof. R. C. H. Heck

J. W. Grant's description of a slide-movement indicator rig, which he evidently considers equivalent to that now regularly applied to the American-Ball engine, has a very considerable element of error; and this error is of the fundamental kind, due not to the difficulties of action at high speed, but to a geometrical defect in the design.

In Fig. 1, herewith, the mechanism is reproduced, with just the proportions of the sketch in the original letter. The fault in the design is made evident when we substitute for the knee-shaped lever

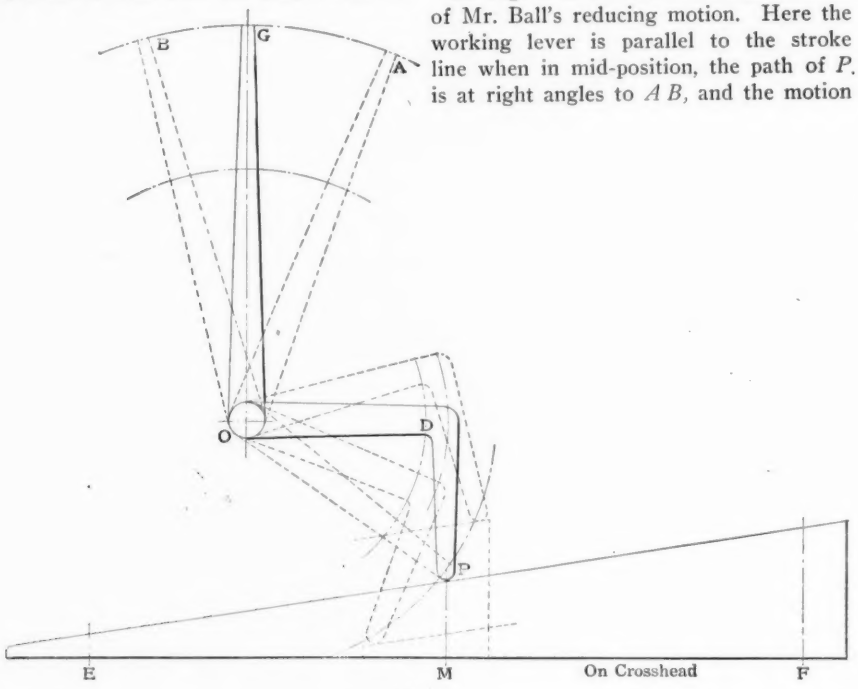


FIG. 1

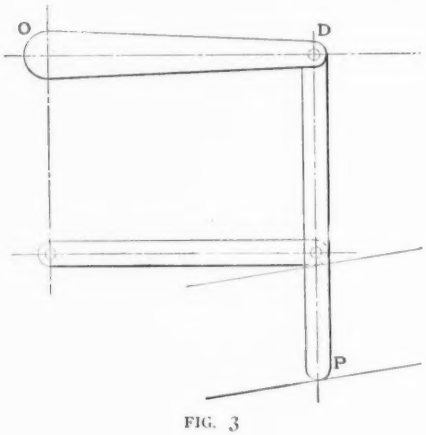


FIG. 3

$O\,D\,P$ the equivalent inclined straight lever $O\,P$, or when we draw the circular-arc path of the center P of the rounded end of the lever. This path P is, in general, oblique to the direction of movement of the slide $E\,F$, or to the horizontal base line of this slide; and since the path is an arc of a circle, the upper part of it has less inclination than the lower part, and that is what causes the trouble.

In Fig. 1 the full-line drawing shows mid-position, with the crosshead at the middle of its stroke. The distances $G\,A$ and $G\,B$ are each equal to half the stroke. By moving the slide $E\,F$, first until E comes to M and then until F comes to M, we get the extreme positions of the oscillating lever. The inequality in the two "halves" of the motion is strikingly apparent.

In Fig. 2 is shown the essential form of Mr. Ball's reducing motion. Here the working lever is parallel to the stroke line when in mid-position, the path of P is at right angles to $A\,B$, and the motion of P is symmetrical. The reduction is correct, the effect of the curvature of the path of P being negligible, even aside from its neutralization by the method of attaching the cord.

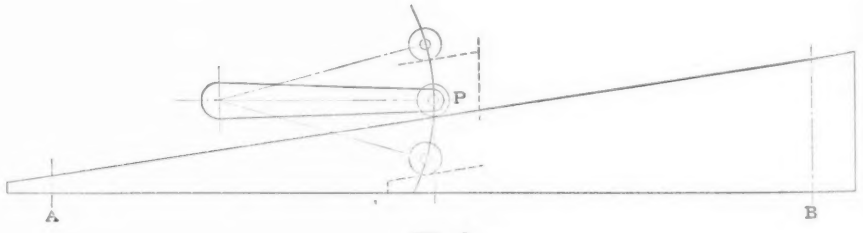


FIG. 2

There are two ways in which Mr. Grant's device can be made correct. One, shown in Fig. 3, is the substitution of a jointed arrangement for the bent lever. By means of a guide link or some equivalent, the rod $D\,P$ is kept vertical or nearly so. This is, of course, merely an addition to the device in Fig. 2, and the added complexity of construction is undesirable.

The other and better way is to put in place of the straight inclined slide $A\,B$ a curve which will give a correct reduction. The method of laying out such a curve is illustrated in Fig. 4. Here $M\,P$ is the working lever, which is to oscillate through the equal angles $P\,M\,a$ and $P\,M\,b$ from mid-position. Divide the path of P, from a to b, into a certain number of equal parts, say eight. Draw through M a line $A\,B$, parallel and equal to the stroke of the crosshead, with M as its middle point, and divide this into the same number (eight) of equal parts, so that the divisions on $A\,B$ and $a\,b$ will correspond.

Then take an angular position of the lever, as $M\,b$, and carry it out to its proper location on the stroke line, at B, where it appears as $B\,E$, parallel to $M\,b$. This is done most easily by drawing a horizontal line $b\,E$, and making the length $b\,E$ equal $M\,B$. The same process is carried out for all the points, and a curve tangent to all the positions of the round end or pin P is the shape of the desired slide.

In effect, we here reverse the relativity of the motion; instead of having the slide move back and forth under the lever, we consider the lever as being moved back and forth above the slide. To the lever is given the intended perfect motion, and it traces on the plane of the slide the profile that will produce this motion.

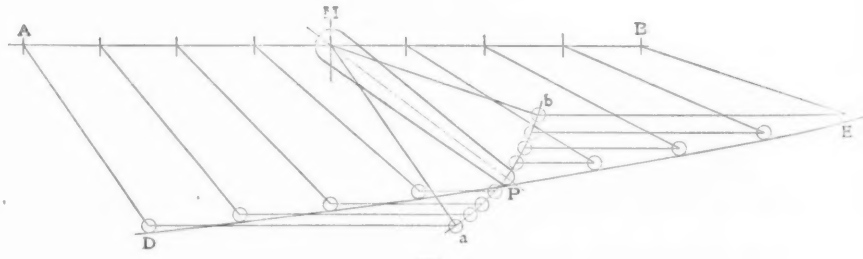


FIG. 4

The drawback to this scheme, besides the difficulty of making the special curved bar, is that there must be a particular curve for every set of proportions. For this reason, the universal device brought out by Mr. Ball, but by no means a novelty, is decidedly preferable.

Reinforced Openings in Boiler Shells

Method of Calculating Manhole Reinforcements Explained in a Thoroughly Practical Manner; Plain Directions Easily Mastered

BY S. F. JETER

Although calculating the proper reinforcement required to restore the original strength to the shell of a boiler or other cylindrical vessel, after having cut an opening such as required for a manhole, is apparently simple, there is probably no other feature of boiler construction about which so little real information exists. Most of the books dealing with the structural details of boilers either skip it entirely or state in a general way that the reinforcing plate should be of equal strength to the metal removed from the shell in cutting the opening. One of the more recent works on boilers says: "The determination of stresses in a manhole ring, even if approximate methods are used, is both difficult and uncertain." This expresses the situation exactly. In fact, there is so much difficulty in attempting to analyze the stresses that are present in this form of construction that there is no probability of any definite knowledge being obtained on the subject until tests on full-sized specimens are made to ascertain just what takes place. The purpose of this article is only to show how manhole reinforcements are generally calculated, give formulas for these calculations and to point out some features about such reinforcements.

Without any attempt at analysis of the stresses, it may be stated in a general way that when an opening of considerable size is cut in the shell of a boiler, and the opening closed by a cover which is not rigidly attached to the shell at the edge of the opening, the stresses introduced in the sheet in the vicinity of the opening, due to internal pressure, tend to deform the shell radially, as well as to produce direct tensile stresses, both circumferentially and lengthwise the shell, and to resist this deformation it is advisable that the reinforcing piece have some depth, so as to resist bending.

That such stresses as pointed out do actually exist is sometimes made evident in boilers equipped with domes, especially if the opening in the shell communicating with the dome has been cut the full size of the dome. In such cases there is some fairly well-defined limit of pressure, at which the boiler may be operated and remain tight; pressures beyond this limit causing distortion of the shell, which results in leaking at the seam where the dome flange is riveted to the shell.

The usual methods of calculating manhole reinforcements ignore the stresses just mentioned and assume that they are the same as if the sheet were flattened out and being pulled apart in a direction at right angles to a plane through the center of the opening and axis of the boiler. Thus in Fig. 1 the solid lines represent the actual position of the sheet in the

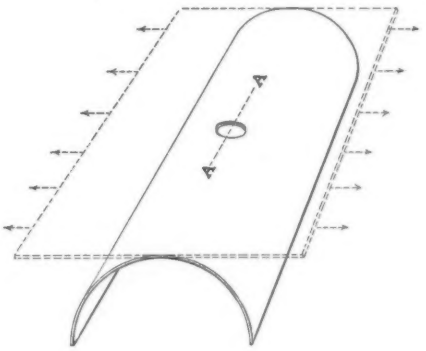


FIG. 1

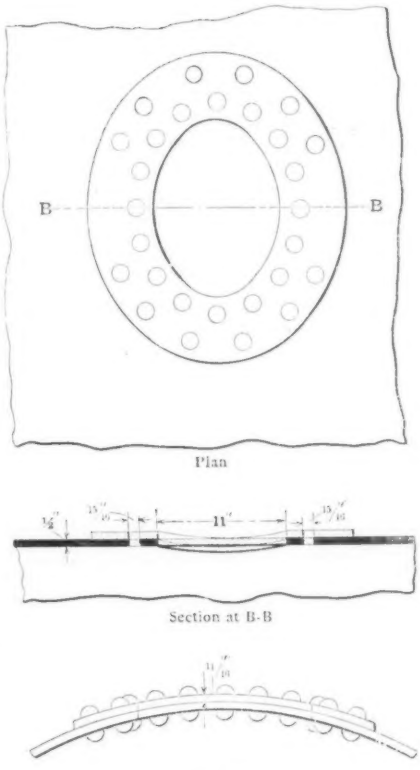


FIG. 2

boiler, while the dotted lines with the arrows show how the stresses are assumed to come on the sheet for the purpose of making the calculations. In elliptical-shaped openings, as used for manholes, the shortest diameter is generally placed lengthwise of the boiler, as illustrated in Fig. 1, for this weakens the shell the least, on account of removing the minimum amount of metal at right angles to the direction of the principal stresses. The usual practice in designing reinforcements is to make them of such proportions that the strength of the metal in tension on a line passing through the center of the opening, parallel to the axis of the boiler, as *A A*, Fig. 1, is the same after the reinforcement has been applied as it was before the opening was cut in the shell, and the reinforcement is double-riveted to the shell. This latter requirement is to insure stiffness, rather than to add strength to resist shearing of the rivets, although it is, of course, necessary to see that the riveting is of sufficient strength to resist the shearing strains brought upon it.

Thus in Fig. 2 the area of metal shown cross-sectioned should be equal to that shown hatched, this latter being the cross-section of the metal removed from the shell. The small hatched portions on each side shown below the blank spaces in the reinforcement represent the metal removed by drilling the rivet holes, which must be added to the amount removed by cutting the opening. The rivets are arranged so that only one on each side need be considered, as is apparent from the plan. To calculate the dimensions of a ring which will have an equivalent section as shown in Fig. 2 proceed as follows: The thickness of the shell would be known, and for present purposes it will be assumed as ½ inch. The diameter of the rivet holes would also be known, for they would be made the same as used in the seams to facilitate construction. Their diameter will be assumed as 15/16 of an inch. The thickness of the reinforcing ring should now be selected, and in the present case it will be taken as 11/16 inch. The only remaining dimension of the ring is its width, which can be obtained as follows, assuming that the ring and shell are of the same tensile strength per square inch:

The area of the metal removed from the shell is

$$11 \times \tfrac{1}{2} = 5.5$$

square inches for the manhole opening, plus

$$2 \times \tfrac{15}{16} \times \tfrac{1}{2} = 0.9375$$

square inch by the rivet holes on each side, or a total of 6.4375 square inches of metal to be replaced by the ring. Since

the ring is assumed to be 11/16 inch in thickness, the width of section of solid metal would have to be such that the width times 11/16 would be equal to assumed to be 42,000 pounds per square inch, and the tensile strength of the reinforcement 55,000 pounds per square inch, then there would be required

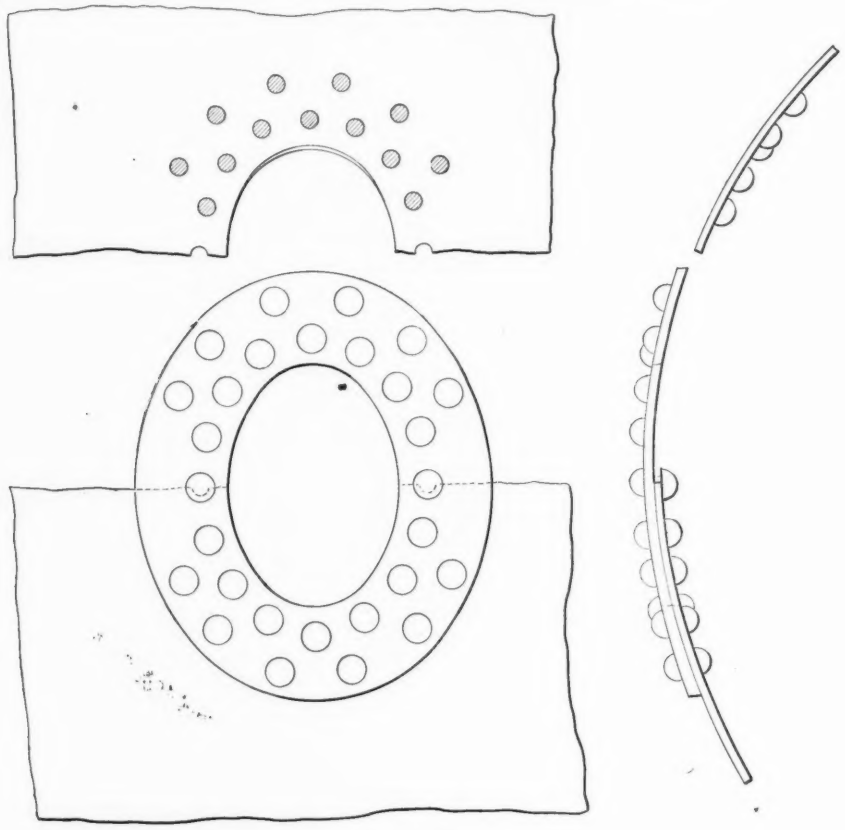


FIG. 3

$\frac{6.4375}{2}$ square inches, the area of metal removed from the shell being divided by 2 on account of only one side of the ring being considered in the calculation, the width of section of metal in the ring would therefore be equal to

$$\frac{6.4375}{2} \div \tfrac{11}{16} = 4.68$$

inches. Now, since this represents the actual width of the section of metal required 15/16 must be added to it, to allow for the rivet hole, as shown in Fig. 2, making the actual width of reinforcement required 5.6175 inches or 5⅝ inches.

The next factor to be considered is the possibility of shearing the rivets which attach the reinforcing ring to the shell and which would result in failure similar to that illustrated in Fig. 3. It is seen from the illustration that one-half of the rivets used in the reinforcement less one (which allows for the two rivets along the line of separation of the plate) must be sheared to cause failure in this manner, and as the strain is considered as being equally divided among the rivets, the rivet area should be properly proportioned to that of the section of the ring, using the tensile strength of the ring and the shearing strength of the rivets in arriving at the correct proportion. Thus, if the shearing strength of the rivets is

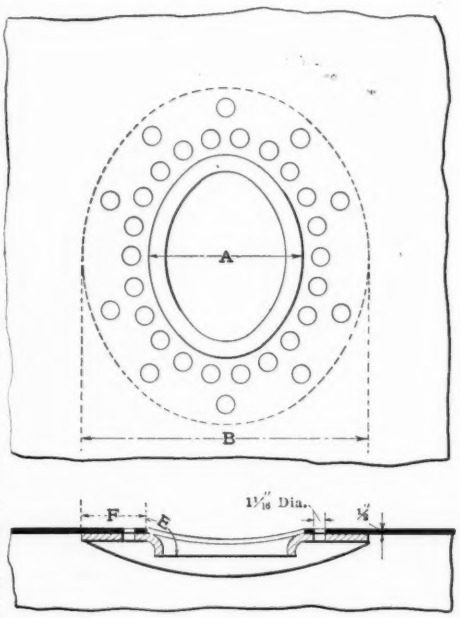


FIG. 4

$$\frac{55,000}{42,000} = 1.31$$

square inches of rivet area for each square inch of area of the section of the reinforcing ring, or in the present case the area of rivets sheared as illustrated in Fig. 3 would have to be

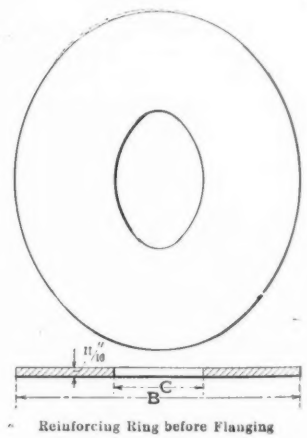
$$1.31 \times 6.4375 = 8.43$$

square inches, or more; and since the area of a 15/16-inch rivet is 0.6903, there would be required

$$\frac{8.43}{0.6903} = 12.2,$$

or thirteen rivets in each half, or $26 + 2 = 28$ rivets or more in the entire ring. The two additional rivets are to cover those on the center line of opening as shown in Figs. 2 and 3. In laying out the rivets on the reinforcing ring it is the custom to place those on the inside row suitable distances apart to permit thorough calking for a tight joint and then adding an outer row to make the connection double-riveted. Often when two whole rows give an excess of rivet area required, every other rivet in the outer row is dropped, as illustrated on the reinforcement shown in Figs. 4 and 5.

Manhole reinforcements of the form shown in Fig. 2 have been placed on boilers and have given good service, but with high pressures there is a tendency toward distortion radially, which adds to the difficulty of keeping the joint tight, and on this account the form of reinforcement generally used on high-pressure work today is shown in Fig. 4, this form being known as the McNeil flanged manhole frame. It is flanged hot from plate of the same quality as the boiler shell. The



Reinforcing Ring before Flanging

flanged edge forming the manhole proper is faced, and in some instances the manhole cover used with this form of reinforcement contains a groove into which the packing is placed, the lip of the reinforcement fitting into the groove. When properly fitted this makes an excellent joint, as it is practically impossible to squeeze or blow the packing out.

To calculate the dimensions of such a reinforcement to replace the metal removed from the shell, it is, of course, necessary to know the dimension *A*, Fig. 4, which in this case is greater than the width of the manhole, the increase being

generally from 3 to 3½ inches more than the manhole opening. It is also necessary to know the dimension C, or width of the opening in the reinforcing plate before flanging. To estimate this dimension on a reinforcement already flanged, the dimension E may be measured with a tape or by rolling a rule around on the surface and then adding this dimension to F to obtain the width of the flange. On account of the drawing down of the metal in the flanging process, it is safest to take dimension E about ½ inch less than it actually measures. After the above dimensions have been obtained, the calculation of the strength of the reinforcement is proceeded with in a manner similar to that for Fig. 2.

Assuming the thickness of the shell and the reinforcing plate and diameter of rivet holes as shown in Fig. 4, the width of opening C as 8 inches and A 14 inches, the following would result in determining dimension B or total width of the reinforcing plate and the number of rivets required:

The metal removed from the shell, in square inches, equals

$$(\tfrac{1}{2} \times A) + (2 \times 1\tfrac{1}{16} \times \tfrac{1}{2}) = 8.0625$$

square inches. Therefore, width B of the reinforcement must be such that

$$[B - (C + 2 \times 1\tfrac{1}{16})]\ \tfrac{11}{16} = 8.0625,$$

or since C equals 8 inches, $B = 21.78$ inches. The area of the rivets on each side of the reinforcement should equal

$$8.0625 \times \frac{55{,}000}{42{,}000} = 10.56$$

square inches, or since the area of a 1 1/16-inch rivet is 0.8866 square inch, the number on each side, exclusive of the ones on the center line, should be

$$\frac{10.56}{0.8866} = 11.92,$$

or twelve rivets. Therefore the layout as shown on Fig. 4 would be all right as regards the rivet strength, since each side contains fifteen rivets.

While it is customary as stated before to have the reinforcing ring and rivets specified of equal strength to the metal removed from the shell, it would doubtless be perfectly safe to design reinforcements for openings as small as manholes, so that the reinforcement would be of the same relative strength when compared with the metal removed as is obtained in the longitudinal seams of the boiler, as compared with the solid plate. To calculate the reinforcement on this basis proceed as follows, using a reinforcement like that shown in Fig. 2 for the sake of simplicity:

Assuming that the boiler in question has a longitudinal joint of 72 per cent. efficiency, then the amount of metal required in the reinforcement would be 72 per cent. of that removed from the shell; and using the same dimensions for thickness of plate and ring and size of manhole opening and rivets as given in Fig. 2, the calculation would result as follows: The metal removed from the shell is 6.4375 square inches, as before. The metal to be replaced by the ring equals

$$6.4375 \times 0.72 = 4.635$$

square inches. The width of the ring =

$$\frac{4.635}{2} \div 1\tfrac{1}{16} = 4.308$$

inches. The rivet area required in each half of the ring =

$$\frac{55{,}000}{42{,}000} \times 4.635 = 6.07$$

square inches, or since the area of a

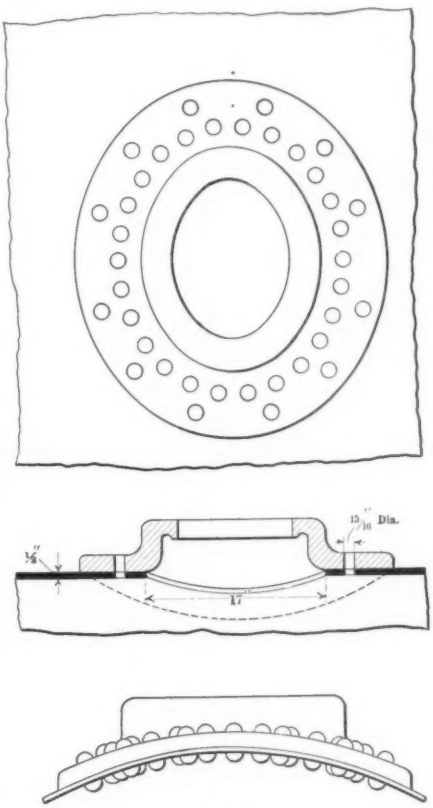


FIG. 5

15/16-inch rivet is 0.6903 square inch there would be required

$$\frac{6.07}{0.6903} = 8.8,$$

or nine rivets in each half of the ring, exclusive of the two on the center line, or twenty rivets in all. A reinforcement equivalent to any joint efficiency may be calculated in a similar manner.

All of the reinforcements considered so far have been assumed to be of metal of equivalent strength to that of the shell. However, they are sometimes made of cast iron, and the calculations have to be modified accordingly.

Fig. 5 illustrates one form of cast-iron reinforcement which is placed on the outside of the shell. It is considered to be the best practice to locate all reinforcements on the inside of the shell, but the form shown in Fig. 5 has one advantage, and this is that in boilers where there is comparatively little room between the top of the shell and the tubes, entrance through the manhole is greatly facilitated when made of this form. This applies especially to boilers of small diameter.

To calculate the proper dimensions of a reinforcement of cast iron as illustrated in Fig. 5, the first step is to decide upon the relative strength of the reinforcing metal and the shell. Gray cast iron when used in boiler construction is generally credited with an ultimate tensile strength of 18,000 pounds per square inch, and assuming the strength of the shell to be 55,000 pounds per square inch, there would be required,

$$\frac{55{,}000}{18{,}000} = 3.05$$

square inches of cast iron to be equal in strength to one square inch of the shell. Therefore, with a 17-inch opening and two 15/16-inch rivet holes in a ½-inch shell, as illustrated in Fig. 5, there would be removed 9.4375 square inches of metal, and to replace this it would require

$$9.4375 \times 3.05 = 28.78$$

square inches of cast iron, or the combined areas of the two sides of the reinforcement shown cross-sectioned in Fig. 5 should be equal to this amount. The rivets required are calculated in the same manner as was done for the other forms of reinforcement, based on the amount of metal removed from the shell. Thus,

$$9.4275 \times \frac{55{,}000}{42{,}000} = 12.35$$

square inches of rivet area on each half, exclusive of the two rivets on the center line, or

$$\frac{12.35}{0.69} = 17.9,$$

or eighteen rivets on each side in addition to the two on the center line, which would make thirty-eight 15/16-inch rivets required in all. If it had been desired to make the cast-iron reinforcement of equal strength to the longitudinal seam, then the metal removed from the shell should be first multiplied by the efficiency of the joint, and the result multiplied by the relative strength between the cast iron and steel plate, and also the same process used in obtaining the rivet strength required, or for an 85 per cent. joint the calculations would be as follows, using the dimensions shown in Fig. 5:

$$9.4375 \times 0.85 = 8.02$$

square inches of equivalent strength to that of the shell plates, which is to be replaced with cast iron, or the total net cross-sectional area of the reinforcing ring should be

$$8.02 \times 3.05 = 24.46$$

square inches, instead of 28.78 square

inches, as before. The rivet strength required would be correspondingly reduced, or

$$8.02 \times \frac{55{,}000}{42{,}000} = 10.5$$

square inches of rivet area required in each half of the ring exclusive of the two rivets on the center line.

To condense the principles for calculating the reinforcements required for manholes as set forth in this article, and make them readily available, it is best to state them in the form of formulas and the following notation is used for this purpose:

T = Tensile strength of shell in pounds per square inch,
T' = Tensile strength of reinforcing metal in pounds per square inch,
t = Thickness of shell in inches,
t' = Thickness of reinforcement in inches,
d = Diameter of rivet holes in reinforcement in inches,
S = Shearing strength of rivets in pounds per square inch,
D = Width lengthwise of the shell of the opening cut in the sheet,
W = Width of each side of the reinforcing ring in inches,
E = Efficiency of the reinforcement as compared with the solid sheet,
N = Total number of rivets required in the reinforcement.

From the following formulas the values of W, N or t' may be calculated when all the other factors are known. If it is desired to design a reinforcement of such proportions that it will be equivalent in strength to the metal removed from the shell, or of 100 per cent. efficiency, the factor E may be omitted, which produces the same result as if it were given a value of unity, corresponding to 100 per cent. When the material of the reinforcement and that of the shell are of the same tensile strength, the factors T and T' cancel out in formulas (1) and (3), and therefore in such cases they may be omitted.

$$W = \frac{t\,T\,E\,(D + 2\,d)}{2\,t'\,T'} + d. \quad (1)$$

$$N = \frac{t\,T\,E\,(D + 2\,d)}{0.3927\,d^2\,S} + 2. \quad (2)$$

$$t' = \frac{t\,T\,E\,(D + 2\,d)}{2\,T'\,(W - d)}. \quad (3)$$

To obtain the efficiency of a manhole reinforcement already constructed, it is only necessary to substitute the correct values in the formulas (4) and (5) and use the one which gives the lowest value for the efficiency, as is done in calculating riveted joints. It should be noted, however, that in the case of reinforcements it is possible to have an efficiency of more than 100 per cent. Formulas (4) and (5) are derived from (1) and (2) by transposition:

$$E = \frac{2\,t'\,T'\,(W - d)}{t\,T\,(D + 2\,d)}. \quad (4)$$

$$E = \frac{0.3927\,d^2\,S\,(N - 2}{t\,T\,(D + 2\,d)}. \quad (5)$$

To make the use of these formulas clear the following examples are given: It is desired to place a manhole reinforcement similar to that shown in Fig. 2 on a manhole which is 12x16 inches in size, the short diameter to be located lengthwise of the shell. The stock available for this reinforcement is bar iron, of ¾ inch thickness and 45,000 pounds tensile strength. It is desired to know what width of bar must be selected to make the reinforcement of 82 per cent. efficiency. The rivets to be used are ¾-inch driven in 13/16-inch holes, and they are assumed to have a shearing strength of 38,000 pounds per square inch. The shell of the boiler is of steel 7/16 inch in thickness and of 52,000 pounds tensile strength per square inch. From this description it is noted that the various letters used in the formulas have the following values:

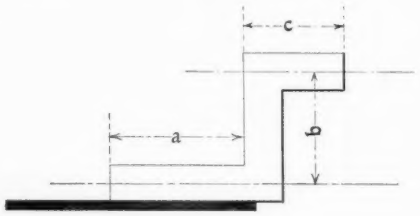


FIG. 6

D = 12 inches,
d = 13/16 or 0.8125 inch,
E = 0.82,
T = 52,000 pounds,
t = 7/16 or 0.4375 inch,
T' = 45,000 pounds,
t' = ¾ or 0.75 inch,
S = 38,000 pounds.

It is desired to find the correct values of W and N and by substituting the above values in formula (1) the following is the result:

$$W = \frac{0.4375 \times 52{,}000 \times 0.82\ (12 + 2 \times 0.8125)}{2 \times 0.75 \times 45{,}000} + 0.8125 = 4.58$$

inches. To determine the number of rivets that would be required in this reinforcement, substitute the values in formula (2) thus,

$$N = \frac{0.4375 \times 52{,}000 \times 0.82\ (12 + 2 \times 0.8125)}{0.3927 \times 0.8125 \times 0.8125 \times 38{,}000} + 2 = 27.8,$$

or twenty-eight rivets. It is, of course, necessary to have the number of rivets used divisible by 2, so they may be laid out symmetrically with respect to the center line lengthwise of the shell. In designing a flanged manhole reinforcement like Fig. 4, formula (3) will be found the most practical to use, for each manufacturer, to facilitate construction, adopts a fixed size of manhole to be used on the boilers which he builds, and this causes the dimensions B and C on the unflanged plate, Fig. 4, to become uniform and therefore the width of flange, or W, is fixed. With the other factors known, formula (3) may be used to determine the different thicknesses of reinforcements required to accommodate the various shell thicknesses. For example, a 14-inch opening, for a 11x15-inch manhole, is cut in the shell of a boiler whose thickness is 9/16 inch and tensile strength 60,000 pounds per square inch, and the reinforcing plates required for this size manhole have been found to be 24 inches across lengthwise of the shell, with an opening 8 inches wide punched out of the center, leaving the dimension corresponding to W in the formulas 8 inches. Assuming that the material of this reinforcing plate is flange steel of 55,000 pounds tensile strength, and that it is to be riveted to the shell by 1-inch rivets driven in 1 1/16-inch holes, the shearing strength of the rivets being taken at 40,000 pounds per square inch. Of what thickness should the reinforcing plate be, if its strength is to be equal to that of the metal removed from the shell? Substituting the above values in formula (3) would give the following results:

$$t' = \frac{0.5625 \times 60{,}000 \times 1\ (14 + 2 \times 1.0625)}{2 \times 55{,}000\ (8 - 1.0625)} = 0.713$$

inch, or the reinforcing plate would have to be 23/32 inch in thickness. The number of rivets required would be found from formula (2) as before:

$$N = \frac{0.5625 \times 60{,}000 \times 1\ (14 + 2 \times 1.0625)}{0.3927 \times 1.0625 \times 1.0625 \times 40{,}000} + 2 = 32.7,$$

or thirty-four rivets required.

To illustrate the use of formulas (4) and (5), assume that a reinforcement similar to that shown in Fig. 2 was placed on a 11x15-inch manhole, the various dimensions and strengths being as follows: Thickness of shell ½ inch; tensile strength of shell, 50,000 pounds per square inch; thickness of reinforcement, 13/16 inch; tensile strength of reinforcement, 48,000 pounds per square inch; width of reinforcement, 5¼ inches; diameter of rivet holes, ⅞ inch; shearing strength of rivets, 38,000 pounds per square inch; number of rivets used in the reinforcement, 28. Substituting these values in formula (4), we get

$$E = \frac{2 \times 0.8125 \times 48{,}000\ (5.25 - 0.875)}{0.5 \times 50{,}000\ (11 + 2 \times 0.875)} = 1.069,$$

or 106.9 per cent.; that is, the reinforcing ring is 6.9 per cent. stronger than the metal removed from the shell.

As regards the strength of the riveting we have from formula (5):

$$E = \frac{0.3927 \times 0.875 \times 0.875 \times 38{,}000\ (28 - 2)}{0.5 \times 50{,}000\ (11 + 2 \times 0.875)} = 0.932,$$

or 93.2 per cent.; that is, the reinforce-

ment as regards the shearing of rivets is 93.2 per cent. as strong as the metal removed from the shell in cutting the opening for the manhole reinforcement and the rivets.

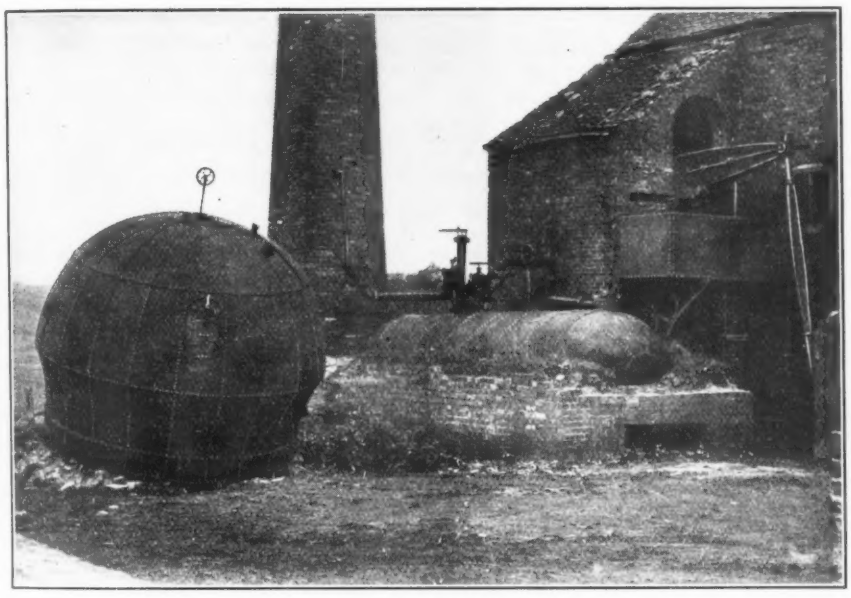
In designing a reinforcement of the shape shown in Fig. 5, it is best first to make a sketch like Fig. 6 to obtain approximate values for dimensions *a*, *b* and *c* and using the sum of these dimensions for a trial value for *W* in formula (3) and determine *t'* and from these values the area of reinforcing metal can be found, after which the dimensions of the reinforcement can be varied at will, the only requirement being to have the area of metal equal to or greater than the amount required.

It is, of course, understood that the formulas given in this article are all based on the assumption that the rivets are so arranged that only one rivet hole on each side of the manhole adds to the section of metal removed from the shell and detracts from the section of metal in the reinforcing ring. To insure that this is the case and allow the two rows of rivets used to be placed reasonably near to each other, it will be found advisable to adopt a layout of rivets on each side of the center of the opening, lengthwise the shell, similar to that shown in Figs. 2, 3, 4 and 5. While the methods of calculating reinforcements as outlined in this article are perfectly safe for openings the size of a manhole, it should be remembered that with a considerable increase in size of openings, numerous factors which have not been considered here would become of such importance as to seriously affect the results, and, as stated at the beginning, practically no definite information exists regarding the problem, when these factors are taken into account.

An Old Haystack Boiler

By H. Maplethorpe

The cut shows a haystack boiler composed of five tiers of 108 plates, there being 21 plates in each tier and one round plate 30 inches diameter at the crown and two plates forming a 4-foot circle for the top of the firebox. There are 21 plates nicely bent all exactly alike to form the bottom ring which rests on a brick foundation. The greater part of these plates have been repaired by some 20 patches. The bottom plates appear to have been 5/8 inch thick and the side plates 3/8 or 7/16 inch thick. The rivets are all 2-inch pitch single riveted throughout. It is provided with a safety valve; a cock is provided for the feed water and it is threaded 1½ inches pipe size and is placed about 3 feet from the bottom on one side.



AN OLD HAYSTACK BOILER

There is a manhole in the top or dome, but there is no sign of gage cocks or water gage. There is not a single square or flat plate and it must have required a great amount of skill and labor to build it. I consider it a boilermaker's masterpiece. It was used for many years to drive a beam engine fitted with parallel motion, cast-iron connecting rod and square cast-iron crank shaft. The flywheel had a square hole about 2 inches larger than shaft and was fitted with 16 wrought-iron and wood wedges. The engine is the same as James Watt's, of 1784, and was used for driving a flour mill. The haystack boiler has been superseded by the well known type of egg-ended boiler of which there are great numbers in this district.

Another feature of this old boiler is that there are no stays in it.—*American Machinist.*

Buying Coal on a B.t.u. Basis

By F. L. Busey

Recent discussion on buying coal on a B.t.u. basis shows a tendency in the right direction, but we must not lose sight of the fact that it is actual performance that really and finally tells the tale. Analyses of samples taken from different parts of the same pile of coal may vary in B.t.u., more especially if the coal happens to be high in ash, or contains slate mixed with the coal.

I have before me an example of this kind, where two tons of coal were loaded from the car onto a wagon and then shoveled onto the boiler-room floor; so it should have been pretty thoroughly mixed. The coal was then divided into three piles, and a sample taken from each, care being used to take a fair sample from all through the pile. Each sample was pulverized, mixed and quartered until a portion containing one thousand grams was left, and after air-drying, this was sent to the chemist. There was a difference of 373 B.t.u. between two of the samples, and the calorimeter a second time verified this difference. Of course, this was probably due to an extra large amount of slate or similar substance being contained in the coal selected as one of the samples.

It is also to be noted that coals on an equal B.t.u. footing do not always give the same economic results when burned under the boiler. An instance of this is shown in the performance of two coals, which I shall designate as "A" and "B." Coal "A" cost 25 cents per ton more than "B," but it was higher in B.t.u., so the cost per B.t.u. was in each case the same.

When it came to actual trial, the equivalent evaporation from and at 212 degrees Fahrenheit for coal "A" was 12 per cent. greater than for "B," and the cost of evaporating 1000 pounds of water was 8 per cent. less for coal "A" than for "B." The cost per pound of fixed carbon was 12 per cent. less for coal "A" than for "B." As to ash and moisture, coal "A" again had the advantage, as the ash for coal "A" was 10.95 per cent., and for "B" 11.88 per cent., the moisture being 7.44 per cent. and 10.24 per cent.

Thus we see that a comparison of these two coals based on their B.t.u. values would not have done justice to coal "A," for after using the two coals or noting the above results one would not hesitate long as to which coal to buy.

An idea as to the amount of energy lost due to condensation of steam in pipes may be derived from the statement that six feet of 6-inch pipe, uncovered, will cause a loss of energy each hour equivalent to that generated by 1 pound of coal under average conditions. A good covering reduces this loss at least 80 per cent.

Refrigerating Experiences

By John H. Ryan

The first bad ammonia leak it was my misfortune to experience will always remain fresh in my mind, on account of its peculiar action on myself. I was repairman in a plant containing thirty-six stands of double-pipe ammonia condensers. They were about the first of this type to be installed and gave considerable trouble from gaskets blowing out. The stands were set on 12-inch centers, which did not give room enough to work around them properly.

One day a gasket blew out on a center connection, and on account of the close quarters and the rigid connection, it took me an entire day to make the repair. The stop valves on that particular stand leaked so much that a candle would not burn in the condenser room. By making frequent trips to the windows I managed to get the job done. I did not notice any effects of the ammonia at the time, but the next day, and for quite a while afterward, my eyebrows were so tender and sore that I thought each hair was being burned around the roots. I have often related this experience to refrigerating engineers and no one ever offered a plausible explanation.

Capacity Increased

About five years afterward I was working in a plant that made counterfeit ice. The real ice crop was short, which created a great demand for the substitute article. The price was more than doubled and the plant capacity was forced up from 230 to 300 tons a day. Water for condensing the ammonia was scarce and the increased capacity put a very heavy load on the condensers. The condensing pressure varied from 240 to 270 pounds per square inch, due to the varying amounts of moisture in the air, for on a muggy day the cooling towers were not very efficient. To hold the condensing pressure as low as possible the cooling water was pumped through the coils at such a rapid rate that the water was heated only 4 degrees when running at the ordinary rate, while the condensing water should have heated up 25 degrees Fahrenheit under proper conditions.

The jar of the water pumping and the vibrations from the compressors, in connection with the high condensing pressure, were enough to cause about fifty ammonia return bends to burst during that season. When one of these bends burst it made a noise that was heard for blocks and crowds of the morbidly curious would gather in the streets to see how many had been killed. The broken fittings usually allowed nothing but hot gas to blow out into the room, but when the break was a bad one, hot gas would come out through the top of the opening, and liquid ammonia back up out of the liquid header and splash over the condensers. We had a safety hood and we became quite expert in shutting off the broken stands. The bends broke at all the different hights on the stands, and formed a practical demonstration of Professor Siebel's theory that superheated ammonia gas gives up its heat very slowly.

There were forty-four stands of double-pipe condensers, with an 1¼-inch pipe inside of the 2-inch pipe. The stands were twelve pipes high and 18 feet long. When the top bend of a stand burst we could hear the roar, but could not see the ammonia escaping because superheated ammonia gas is perfectly invisible. The only way to find the break was by going along in front of the stands until the blowing gas could be felt, when that particular stand was shut off.

The liquid-ammonia stop valve was always shut first, because if the top valve was shut there would be a clear field for liquid to splash out through the break, and then it required pure nerve to crawl through it and shut the stop valve. The escaping gas always came out transparent when any of the top eight pipes were open, but when the fourth pipe from the bottom was open the escaping gas could just be noticed, as it was a little smoky, and the gas coming out through a break on the bottom pipe of a stand would look like a blue mist.

From these breaks it looked as if twelve pipe stands of condensers were not high enough for can-plant work where a distilled-water forecooler was used, because it took eight pipes to take out the superheat and left only four pipes to take out the latent heat so that the ammonia gas could condense into a liquid. It is not consistent engineering to expect four pipes to take out twice as much heat as eight pipes will; the proportions should be reversed. We blew away thousands of dollars worth of ammonia that summer and the foregoing were all the useful things we learned.

Physical Suffering

One day a bottom bend burst and the break opened in such a way that liquid ammonia was blown out on the bottom stop valve. To shut this valve I was forced to crawl through about 10 feet of blue gas, and while it took only a few seconds to close the valve, I experienced severe burning sensations. We had another break about fifteen minutes afterward, which also happened to be on a lower pipe, and I experienced the same physical effects. The other men who shut off any of these breaks always experienced the same sensations when going through an ammonia fog.

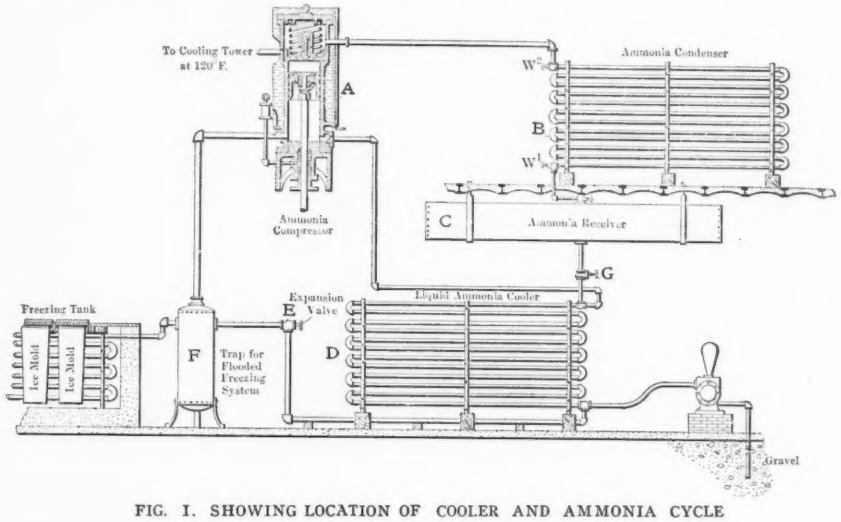


FIG. 1. SHOWING LOCATION OF COOLER AND AMMONIA CYCLE

We read everywhere that ammonia gas is lighter than air and that the safest place is near the floor when ammonia is escaping. This statement needs to be qualified somewhat, as my experience has been that the ammonia gas will stay close to the floor as long as it is visible, and it rises only when it becomes superheated by contact with warm air. I have known ammonia to settle close to the floor, in cold rooms, when it was perfectly safe to walk through the ammonia fog.

An Experiment

In a certain ice factory cooling water was very scarce and cooling towers were used. They had enough well water to make up the evaporation on the towers, and the question arose as to how this small amount of cold water could be used to the best advantage. Mixing it in with the general supply from the towers did not appear to be doing a great amount of good, because the proportion was so small that the difference in head pressure could not be noticed, and the liquid ammonia appeared to be just as hot with the well water in use as without it. As an experiment, a liquid-ammonia cooler was put in, which consisted of two stands of old double-pipe brine coolers. They were twelve pipes high and 18 feet long, with a 2-inch pipe inside of the 3-inch pipe.

As shown in Fig. 1, the cooler was placed on the liquid line between the ammonia condensers and the freezing system. The entire ammonia cycle for this plant

is also shown. At *A* is shown the single-acting compressor, at *B* the double-pipe ammonia condensers, the hot ammonia gas from the compressors entering at the top and passing downward, while the cooling water enters at W^1 and leaves at W^2.

This style of condenser is very economical in the use of water, when enough stands are installed, as the water will take up three times as much heat as those of the atmospheric type. The liquid ammonia drops from the condensers into the receiver *C*, and in warm weather the temperature of the liquid ammonia in the receiver varied between 85 and 100 degrees Fahrenheit. When this warm liquid was fed to the freezing system, its first duty was to cool itself from this high temperature down to the brine temperature at 12 degrees Fahrenheit. This lost work was a large per cent. of the total useful work in the ammonia and, to try and cut it down, the liquid cooler at *D* was put in. In this cooler the liquid ammonia and the cooling water traveled in opposite directions, and exchanged some of their heat. The well water was obtained at a temperature of 50 degrees Fahrenheit, and the cooler cooled the liquid ammonia down to the same temperature. This was doing excellent work and the capacity of the plant was increased quite a lot. In fact, it made it possible for them to pull as much ice in July, with the outside temperature at 90 degrees Fahrenheit, as they could pull in the winter when filling their storehouses.

The expansion valve is shown at *E*, and the expansion separator or accumulator is shown at *F*. This arrangement keeps the coils in the freezing tank full of liquid, and makes them more efficient, and any gas coming through the expansion valves goes right back to the compressors. The arrangement does not look practical, but the danger of short-circuiting is only a theoretical one, as there is no more fear of the liquid coming out of the accumulator than there is of water being lifted by a pump whose suction pipe does not reach the water.

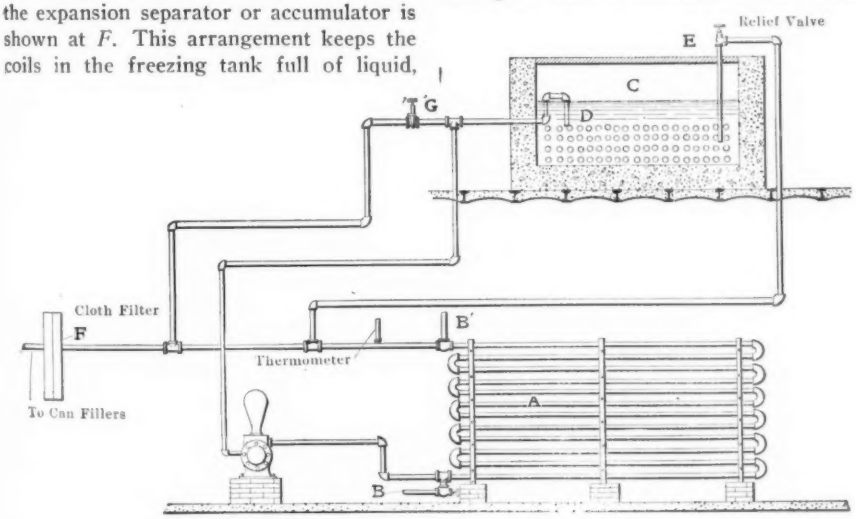


FIG. 2. HOW THE OLD COOLER WAS UTILIZED

The well water is pumped through the liquid-ammonia cooler, where it gets warmed up from 50 degrees Fahrenheit to 80 degrees Fahrenheit, and from there goes to the water jackets of the ammonia compressors. The lower ends of the jackets were filled with cement up to a point where it was calculated that the temperature of the ammonia being compressed would be higher than the temperature of the entering water. The overflow from the jackets ran into the suction of the cooling-tower pump, because it was cooler than the water coming off the steam condensers, and if there was any water to waste they always wasted it as hot as possible.

Why Breaks Occurred

The liquid cooler *D*, Fig. 1, was improvised out of an old brine cooler, as before stated, which, when designed, was not expected ever to have to withstand more than 45 pounds pressure per square inch. The consequence was that when the head pressure ran up above 220 pounds, the bends would split. As it was a liquid that was escaping, it made no roar similar to a gas break.

The cooler was set on the insulation space between two freezing tanks, and an engineer was standing close to the cooler when the first break occurred. He ran over and shut the valve *G*, Fig. 1, and then turned the expansion valve *E* wide open. The compressors being in operation pumped the pressure off the low side in a few minutes, so that there was only a small loss of ammonia. The managers were so convinced of the worth of the liquid cooler that they bought a new one designed for the pressures it would have to resist.

Disposal of the Old Cooler

When the new cooler was installed the old one was moved up into the engine room, where it was always in view, and every once in a while someone would suggest a way of putting it to some useful purpose within its limit of strength, and finally they decided to put it on the sweet-water line between the storage tank and the cloth filters going to the cans.

The arrangement was about as shown in Fig. 2. The cooler *A* was cooled by direct expansion of ammonia, fed by the valve *B*, and the ammonia gas returned to the compressors by the pipe *B'*. The sweet-water tank *C* had return ammonia-gas coils in the bottom, and the water to the pump left the tank through the pipe *D*. The end of the pipe inside the tank was connected up as shown, to prevent air from getting into the water when the water level was low; when the top of the water was even with the horizontal pipe air would be drawn into the pipe and it made white ice.

With the present arrangement the first bit of air that enters the pipe will break the vacuum and no more water will run out until the level rises up to the top of the inverted loop. This was arranged this way when the cans were filled by gravity, and it prevented the ice pullers from rushing, as they soon learned that when the water stopped there would be no more for an hour. With the pump, the rate of filling was under the engineer's control, by simply regulating the speed of the pump. Any time that the can fillers were not working the cooled distilled water was pumped back into the storage tank through the relief valve *E*. The valve *G* was for gravity feed when the cooler was cut out for any reason.

Pipe Froze

The first benefit noticed from the use of the cooler was in the increased amount of dirt that the cloths in the filter *F* took out. It appears that iron solutions are precipitated when the temperature gets below 35 degrees Fahrenheit. With the cooler in use, it was discovered that they could get capacity with a higher back pressure of ammonia, which increased the compressor efficiency. Naturally, when it was seen that there was any merit in the rig, they worked it to death, with the usual consequence of something giving way. At any time that the ammonia was frosting through the storage tank, the water in the tank would be colder, and then the water pipe in the cooler would ice up and gradually close off altogether. The operating force were not aware of this, and one day it froze up solid and burst the 2-inch inner pipe.

When the pipe split open it let the ammonia come in contact with the ice inside the pipe and the ice absorbed the ammonia, with its consequent generation of heat. This heat melted the ice and in a couple of minutes the coils were clear of ice and the pump started up again. It blew ammonia into the distilled water for about fifteen minutes before the engineer noticed the rattling noise in the cooler, and cut it out for investigation. The cans that were filled while this was going on were pulled out and the water sold, as aqua ammonia, to a dyehouse.

Practical Letters from Practical Men

Don't Bother About the Style, but Write Just What You Think, Know or Want to Know About Your Work, and Help Each Other

WE PAY FOR USEFUL IDEAS

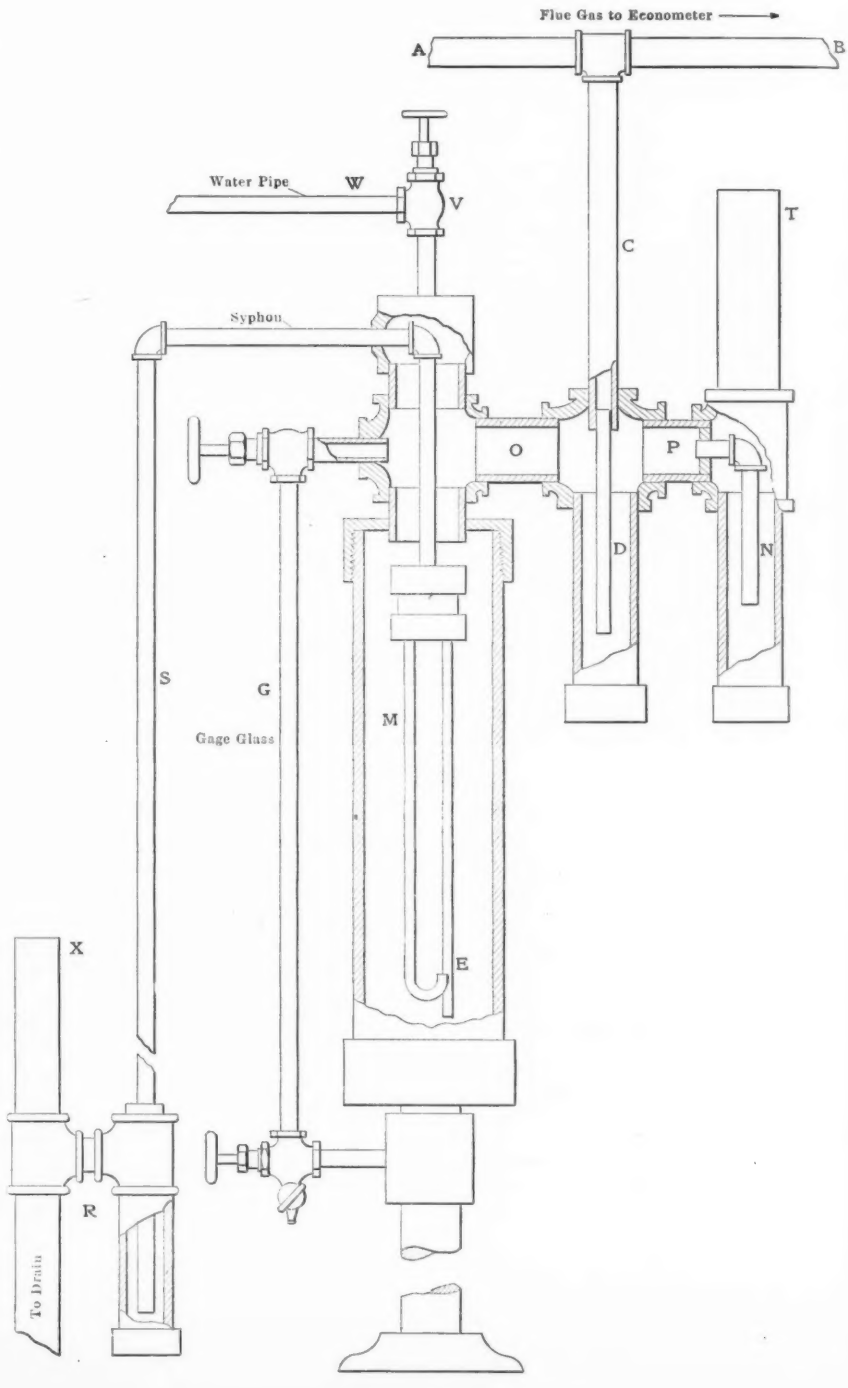
A Pump without Moving Parts

The pump shown herewith was designed for use with a flue-gas econometer. It has no moving parts, and is entirely automatic in action. The function of this pump is to remove the gas from the piping, so that the econometer will have a fresh supply for each analysis. The necessity for this device arose from the fact that, to be of most value, the econometer should be placed in front of the boilers, for convenient observation by the fireman. This required a comparatively long pipe line which would retain a large amount of old gas.

The pump is composed principally of pipes and fittings. The chamber *M* is alternately filled with water and gas. The pipe *S* has a divided end; one division turns upward in the chamber *M* and serves as a siphon to empty the chamber. The water pipe *W* and the valve *V* admit water to the chamber. The gage glass *G* shows the working of the pump by indicating the rise and fall of the water in it. At *D* and *N* are induction and eduction water seals, respectively. The former has the longer seal when discharging, and the shorter when the chamber *M* is filling with gas. At *R* is a drain, which may or may not be water-sealed. The pipe *C* connects the pump with the pipe *A B*, at or near the connection with the econometer. The pipes *T* and *X* are placed on the apparatus to keep the water from splashing on the floor. The gas is drawn from the flue through the pipe *A B* to the econometer. The chambers or pipe *M* has a volume equal to the volume of the piping from the flue.

The operation of the pump is as follows: The water is turned on, and allowed to flow slowly into the chamber *M* until it is full enough to start the siphon *S*, which quickly empties the chamber *M*. As the water falls in *M* it draws the flue gas from *A B* through the pipe *C*, through the water seal *D*, and over into the chamber *M*. When the water falls to the point *E*, in *M*, the gas enters at *E* and breaks the siphon. The water then refills the chamber *M*, forcing the gas through the piping *O* and *P*, through the water seal in *N*, and out to the atmosphere through the pipe *T*. When the gas has been entirely discharged, the chamber *M* is full of water, when the siphon starts again and the process is repeated, thus securing a fresh supply of gas for each analysis.

It is interesting to note that the water seals have their supply of water kept up by being flushed each time the pump discharges a cylinderful of gas, so that the pump operates continuously and without attention.



DETAILS OF PUMP WITHOUT MOVING PARTS

The device has been in satisfactory operation for several years in the boiler room of the Elgin watch factory.

W. W. PARKER.

Chicago, Ill.

Simple Oiling System

Herewith is shown a simple oiling system which I designed about five years ago, and which has been in successful operation ever since. There are no patents on it.

The operation of this device depends entirely on the well known fact that oil and water will not mix, and the only precaution necessary is to see that the outlet pipe from the tanks is kept full of oil at all times. In the drawing are shown two duplicate tanks containing the oil, although one would be sufficient, but I keep two so that the filtered oil from the engines can be poured into one daily, without interruption to the service.

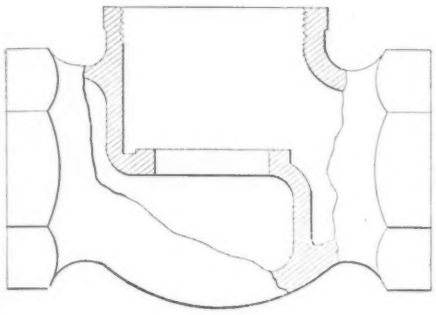


FIG. 1

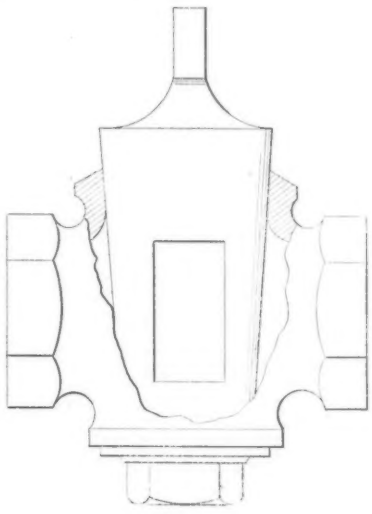


FIG. 2

The operation of the device is as follows: To fill either tank, the outlet valves *A* or *B* and *C* and *D* are closed, and the drain valves *E* or *F* and the air cocks are opened, permitting the water to drain to the sewer or into a bucket, until the water shows at the bottom of the gage glass on the end of the tank. The sewer valves are then closed, and valves *G*, *H* or *J* opened and the filtered or barrel oil poured in through the filling funnel.

After the tank is filled, or partly so, the valves *G*, *H* or *J* are closed and the water-supply valves *C* or *D* are opened until the oil is level in the tank and shows at the air cock, which is then closed and the supply valves *A* or *B* opened. Should air accumulate between the tank and valve *A* or *B*, the faucet below may be opened or used to draw oil for other purposes.

Should it be desired to use this device for heavy cylinder oil, if located in a cold place, it will be necessary to use a heating coil in the tank with a small amount of exhaust steam passing through it.

Any type of tank can be used, or an oil barrel with a moderate water pressure, but the ordinary kitchen-range water tank is what I am using. Those of 60 gallons capacity will answer very well, and they are inexpensive. The oil may be used from tanks direct to the oil cups on bearings or fed to a reservoir on the engines.

J. S. HILL.

Bethesda, Md.

Sugar in Boilers

On page 730 of the November 3 number will be found the statement that the only detrimental effect caused by sugar in boilers is its tendency to cause foaming; otherwise, its presence is beneficial.

Some years ago I read in a technical journal that while sugar will undoubtedly remove scale, it will also dissolve nearly as much iron, and one of the problems in carrying sugar in iron ships was on account of its corrosive qualities; also, that any iron parts of sugar machinery are dissolved, the sugar attacking the iron with extreme rapidity, and when in contact with air is precipitated as ferric oxide. The sugar is then free to attack the iron again, and a small quantity of sugar will thus corrode deeply into a plate of iron.

On page 745 of the same number, F. C. Tryon states that the entire supply of coal of the world, including all developed and undeveloped sources of supply, will probably be consumed in hundreds instead of thousands of years. Now, it runs in my mind that I read in an editorial in POWER AND THE ENGINEER some time ago that the soft coal in sight would probably last for thousands of years.

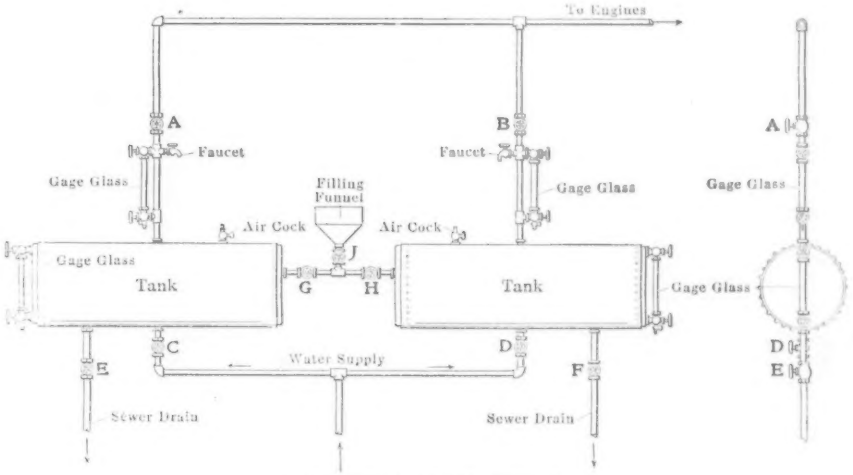
In the same article it is stated that but from 6 to 10 per cent. of the heat energy of steam is utilized. This is only in isolated cases, for in most plants the engine exhausts are used for heating water, or for some other purpose, and in many plants about the only loss of heat of any considerable quantity is what escapes from the stack after it has passed through an economizer.

JAMES E. NOBLE.

Toronto, Can.

Blowoff Valves

Figs. 1 and 2 show two kinds of blowoff valve; Fig. 1 represents the globe blowoff valve and Fig. 2 the stop or asbestos-packed valve. I have found that the valve shown in Fig. 1 is unfit for use where the water scales. A flat seat is worse than the valve with the common taper seat used on plain globe blowoff valves. The type of valve shown in Fig. 2 closes in a manner that makes it almost impossible for scale to cut the seat.



MR. HILL'S OILING SYSTEM

A plan the writer used when the so-called disk valve first came out was as follows: Before placing the disk in the valve, a washer of thin rubber 1/32 inch thick was cut out and placed in the disk cup. The disk was then placed in the cup and screwed up tight. This prevented the breaking of the disk and made it last much longer.

C. R. MCGAHEY.

Lynchburg, Va.

Operating Generators Together

I have two generators, one 115-kilowatt and one 150-kilowatt, both operating at 150 revolutions per minute, belted to a jack shaft. We will call the 115-kilowatt generator No. 1 and the 150-kilowatt generator No. 2. The driving pulley for No. 1 generator is 68⅝ inches in diameter and that for No. 2 is 68⅞ inches in diameter.

With the jack shaft turning at 300 revolutions per minute, No. 1 machine ran at 888 revolutions and No. 2 at 903. When we put them together their voltage remained constant, but with no load on the machines the amperage would show a total of about 90 amperes.

At another time we had two 115-kilowatt generators running from the same jack shaft with driving pulleys 68⅝ and 68⅞ inches, both pulleys on the generators being 22½ inches in diameter. The clutch to No. 2 generator had been broken, and we drove wooden wedges under the knuckles to make it carry its load. While the clutch was in this condition one of the generators was replaced by a 150-kilowatt machine, and ran together all night; but when the parts to the broken clutch came and were put on, there was trouble, as the 150-kilowatt machine would do all the work, besides making a motor of the other machine. The pulley of the 150-kilowatt machine was lagged with sailcloth, wound around the face of the pulley, when the generators ran together without trouble.

JOHN F. DAVIS.

Clinton, Mass.

Power Plant Output Records

In the December 8 number, in an editorial under the above caption, it is stated that usually no means for keeping accurate records of coal and water consumption are provided in the ordinary power plant, and that the average operating economy of the outfit is usually based on the coal company's bills and some haphazard reading of switchboard instruments. It is also stated that "no matter what other instruments are or are not provided, every station should have in regular service reliable means for measuring the coal fired under the boilers and the water fed to them."

Power-plant managers are fast coming to realize that these statements are all too true. Lack of an accurate, reliable means for automatically weighing and recording the feed water has, until recently, been largely responsible for this condition of affairs. A largely increasing number of plants are, however, installing water-weighing apparatus, and some interesting results are being attained. As an example the Dow Chemical Company, of Midland, Mich., may be cited. At this plant a water-weighing device operating on the principle of weighing by means of a column of water of standard hight is employed. A daily record is kept of the water fed to the boilers, and of the car invoices of coal burned.

Recently the question of coal contract for the ensuing year came up, and among the coals considered were three brands of slack, all from the same coalfield, and all at the same price per ton, $1.60, delivered at the plant. There appeared to be no choice, so the manager of the plant bought 100 cars of each kind, enough to run the plant for about three weeks with each coal. The results were as follows: Coal A evaporated 1000 pounds of water for 12.6 cents; coal B for 11.3 cents; coal C, 11.1 cents.

The plant has 1200 horsepower of boilers, and in twenty-four hours evaporates about 2,250,000 pounds of water. The difference between coals C and A is 1½ cents for each 1000 pounds of water evaporated, making a net saving in coal of $33.60 per day.

Such results are certainly worth consideration, even by the manager, who, as quoted in the editorial referred to, will freely admit that the necessary recording instruments "should have been put in, but—."

GEORGE B. WILCOX.

Saginaw, Mich.

Putting an Engine on Center

There have been several letters published recently dealing with putting an engine on the center, and proposing methods which, while correct, as far as results go, are easier described than done, if the engine is a heavy one. My way does not require as many movements as those I have read. It is as follows:

Draw a line direct from the center of the crank pin through the center of the shaft and continue it out over the edge of the crank with the engine on the center and level with a spirit level. If the connecting rod interferes when on the extreme out-stroke draw another line parallel with the first line and level. Then make all marks with a lead pencil so they may be erased without defacing the crank or other parts of the engine.

L. JOHNSON.

Exeter, N. H.

Poor Steam Piping

Fig. 1 shows a system of piping installed in a first-class steam plant. In the first place, the large reduction tee made

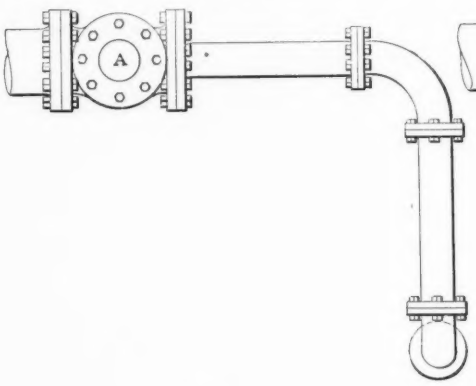


FIG. 1

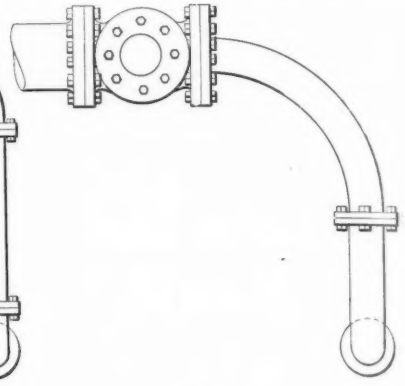


FIG. 2

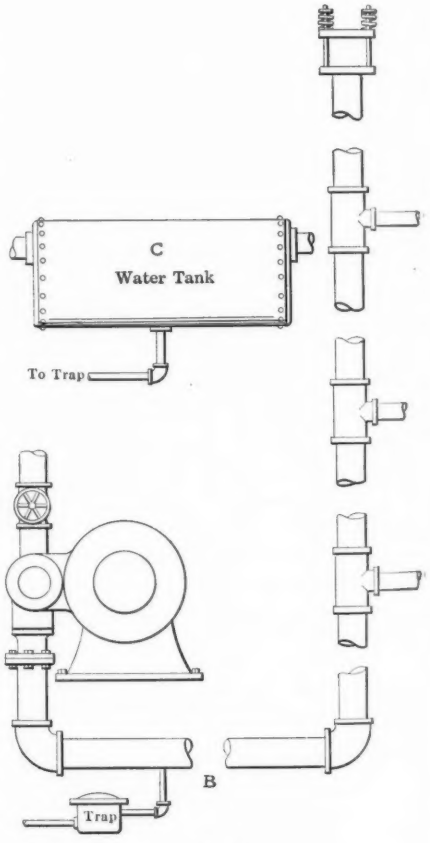


FIG. 3

a good water trap, as the side outlet was blanked off as shown at *A*. The tee is not placed right, but placed as it is, it should have been supplied with a drain pipe. This pipe feeds a compound engine which has already had two new low-pressure cylinder heads, the effect of slugs of water. Fig. 2 shows a better way of doing the piping, and the cost would not be any greater.

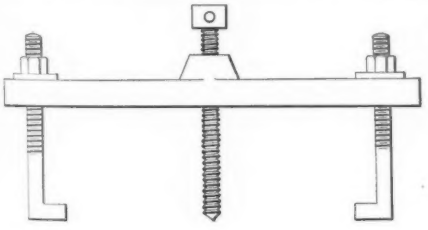
In another instance the exhaust from a high-speed piston-valve engine was working directly into the heating system, having a relief valve which regulated the amounts of back pressure the engine was to carry. The heating system consisted of one large outlet pipe, having smaller pipes leading to coils on the various floors of the building, all connected, as shown in Fig. 2, and connected to a small trap which is supposed to relieve the piping system of any accumulation of water. The trap was connected in about the usual manner. The engine was provided with a good set of relief valves to care for an overdose of water. One day the engine had to be shut down after making the usual start in the morning and, upon restarting, the engine went to pieces, due to water in the cylinder. This was because the safety devices were so placed that they could not be of value to the system. A small steam trap, capable of caring for the system nicely when running under normal conditions, had been installed, but in this case the water came back to the engine and blew the cylinder head out. Had this system been erected with a tank placed in the line at *B*, piped as shown at *C*, the wreck would not have occurred, and the operating engineer would have held his job. This tank would care for an over amount of water and the trap would carry it off when starting up.

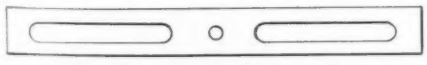
C. R. McGAHEY.

Lynchburg, Va.

Drawing a Tight Key

I have seen keys drawn in a manner which resulted in great damage to the shaft, key and wheel. Therefore, I rigged up a simple device for removing keys without injury to anything. A heavy piece of 5/8x2½-inch flat iron was procured, in the center of which a hub was





FOR DRAWING A TIGHT KEY

welded on. This hub was drilled and tapped to fit a ¾-inch screw spindle. To each side of the hub a longitudinal slot was cut to within 1 inch of each end, and the same distance from the hub. Then two bolts were made with a long thread on one end and a solid angle at the other end. The device is shown in the accompanying illustration.

The spindle is placed in the center of the shaft, with one hook behind the key and the other behind a spoke of the wheel. The screw spindle is then screwed in and the key easily removed.

C. H. BAAS, JR.

New York City.

Natural Gas under Boilers

In our plant, which contains six boilers, we burn natural gas, using the Buckeye burner, four of the boilers having been in continual service more than four years. The burners are capable of maintaining a blue flame under light load, and a light straw-colored flame at maximum load, and at all times they give nearly perfect combustion.

The average consumption for ten days in September of this year, taking the first and last five days of the month, of three boilers furnishing steam to two 500-horsepower engines, running slightly below their indicated horsepower, besides operating the boiler-feed pumps, a deep-well pump, a steam hammer, etc., the three boilers being rated at 250 horsepower each, was 27.95 cubic feet per boiler horsepower. Our daily and monthly reports show that natural gas is cheaper than coal, if proper combustion is maintained. We rarely exceed an average of 30,000 cubic feet per hour, and have no trouble at all in getting the desired mixture of gas and air, as the burner is controlled by one valve, the air being drawn through the ashpit doors, which are removed.

The stack draft is sufficient for from 4 to 8 ounces of gas, but if necessary to crowd them to handle the peak load for a few hours, means are provided to supply them with just enough additional air to maintain proper combustion.

We have only repaired our burners once in the last four years and have no burned nozzles or clogging up of air or gas passages. The side walls and top of the furnaces, with the blue flame, are a dark cherry red, and with the light straw color, almost incandescent.

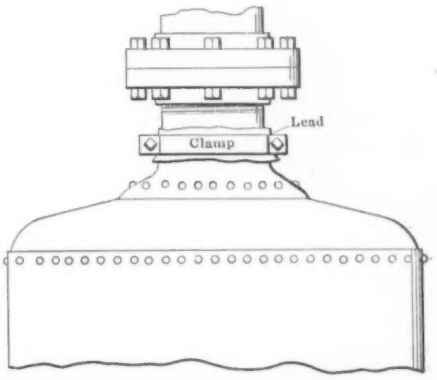
W. D. RANNEY.

Columbus, O.

Stopping a Leak in the Steam Receiver

Some time ago I found it necessary to stop a leak in the upper collar of a large steam receiver over our main engine. Finding that calking did very little good, as there was considerable vibration in the steam pipe, I put a piece of asbestos paper next to the joint, then a piece of thin sheet lead, and a clamp made out of 3/8x2-inch iron. The clamp was tightened with two bolts and nuts. The clamp was then driven down solid on the collar.

For a time this answered very well, but a few days ago it started to leak a little around the bottom of the clamp. Thinking I could stop the leak, I was surprised when I touched the iron clamp with a calking tool, to see an electric spark jump from the clamp to the tool. I repeated this a number of times, perhaps a dozen, and every time the tool touched the clamp an electric spark was produced; I then touched the lead and after that could not get any sparks.



REPAIRING LEAK IN A STEAM RECEIVER

Can any reader tell me the cause? My own theory is, that the lead, which was insulated from the pipe and receiver by the asbestos paper, acted as a kind of storage cell; perhaps getting its charge from our 44-inch belt (which was 15 feet from the clamp) through the metallic connections of the flywheel, crank shaft, engine bed, cylinder and receiver.

GEORGE A. RAYMANT.

La Grande, Ore.

Why Some Engineers Do Not Read

It is a deplorable fact that many engineers do not read engineering papers, especially those of a technical nature. As I have been questioned recently as to my opinion of what articles appearing in engineering papers are of most interest to engineers, I will say, without hesitation, that the practical letters are best of all.

There are many reasons why most engineers do not read technical papers, but one of the principal reasons, and I speak from experience, is that they do not talk shop in an attractive or interesting manner. Algebraic formulas and elaborate data do not satisfy, but have a tendency to mystify, and consequently interest is lost before reading many paragraphs.

Simple talks of actual happenings of the power department, telling just how some particular job was done, and the consequences of not properly doing others, will attract and hold the attention of engineers more than technical articles of whatever nature.

The first articles appearing in most engineering papers are usually deeply interesting because they describe condi-

tions in other plants, and any new thing or condition, if properly illustrated and described, will hold the attention of engineers every time.

Submitting almost perfect diagrams for criticism, telling the best way to fill a lubricator and wipe an engine, or why the chief engineer should help clean the windows do not teach us anything, but rather promote the habit of skipping some things, which causes the reader to lose interest in his engineering paper.

W. D. RANNEY.

Columbus, O.

Repairing a Broken Valve Stem

The breaking of the crank-end exhaust-valve stem on a Corliss engine placed the engineer in an annoying situation. The valve stem broke at about 5 p.m. and, as the cotton mill shut down at 6 p.m., only one hour was lost. It was very

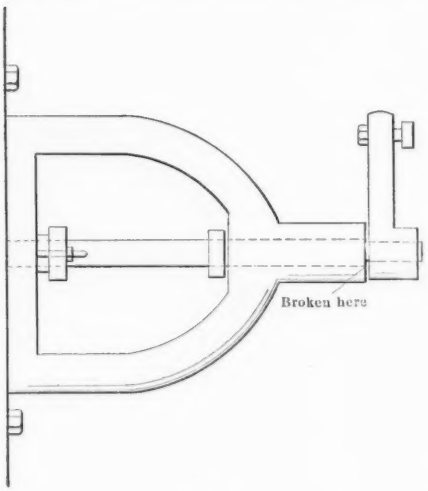


FIG. 1

essential, however, that the plant be in operation the next morning.

As the stem broke off close up to the shoulder, as shown in Fig. 1, in order to make the repair the whole valve had to be taken out, as the valve and stem were cast together. The valve was put in a lathe and a hole drilled into the shoulder, as shown at *A*, Fig. 2. A steel plug *B*

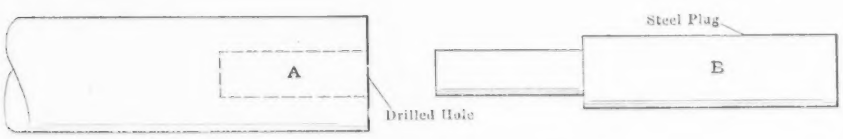


FIG. 2. ENLARGED DETAIL OF REPAIR

was turned to fit this hole, and driven in. Three holes were then drilled through the two pieces and pins driven in to make a solid job of it. The end of the plug was then turned off to the size of the broken valve-stem end, and thus a neat and substantial repair was made.

The valve was then reset and the engine was ready to start at the usual time the next morning.

W. L. WHITMARSH.

Phenix, R. I.

High Pressure Flanges

The use of heavy or high-pressure flanges on steam and water lines does not

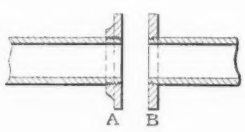


FIG. 1

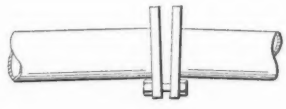


FIG. 2

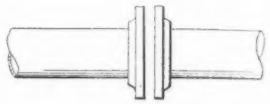


FIG. 3

necessarily mean that they are used for the sole reason that high pressure is to be carried on the line. The heavy flange has another advantage; it has more depth of thread, gets more bearing on the pipe and, consequently, makes a better joint and is not so likely to leak. For instance, in Fig. 1, if the two flanges *A* and *B* were put on with the same tension, flange *A* would hold better than *B*, because it has more thread surface.

Another example is shown in Fig. 2. In some cases of heavy piping, the joints are pulled together, as shown by the bolt. When thin flanges are pulled up in this manner they are more than likely to spring and cause a leak in the flange and pipe when the pressure comes on the line. The heavy flange will stand more abuse than the light one without causing the joint to leak.

Pipe-line flanges should come up parallel as shown in Fig. 3, so the strain will be even on the pipe and on the flanges. Many joints leak on one side of the flange from this cause, due to the strain, which is often increased by the expansion of the line, making the condition worse.

C. R. MCGAHEY.

Lynchburg, Va.

Misreading Steam Gages

A few days ago I stepped into a boiler room to look around, just as they were about to cut in a boiler. The steam gage was so covered with dust I could hardly see the pointer, but it appeared to have 125 pounds pressure, same as the other boilers. When the water tender cut the boiler in there was a loud report and I saw the gage hand swing around from about 15 to 130 pounds and then back to 125. When I looked around the room the last man was just going through a window.

By painting the short end of the gage hand white, or the same color as the face, there is no chance of misreading the gage, and the gage can be read more quickly at a distance or in a poor light.

One morning some years ago I came on duty as usual at 6:30 (the night fireman left at 6), and thought I had 80 pounds of steam. I went into the engine room and started up the lighting engine. When I looked at the engine-room gage, it showed 190 pounds. I hurried into the boiler room and that gage also indicated 190 pounds. I started to pull the fire, but just then the safety valve blew off with a loud bang. Since then I have always painted the short end of the hand the same color as the face of both vacuum and pressure gages. I have also found that by painting the face black and the letters white they are much more easily read.

A. BALDWIN.

Seattle, Wash.

The Braces Broken

I once took charge of a plant at a coal mine. It was equipped with a plain cylinder boiler and a return-tubular boiler. I started to work on Wednesday.

The following Sunday I opened the cylinder boiler to clean it, and found all the brasses of the front head broken, which condition they had been in a long time, as the ends of the rods had drawn apart enough so scale and dirt had filled the break. If I had not hit them with a hammer, I should not have known they were broken. The front head was bulged out more than usual.

I called the superintendent's attention to the matter and nearly lost my job because I insisted on having the braces repaired before I fired the boiler again. His argument was that it had run so long in that condition it could continue, and he did not see any use in having the braces put in.

That experience taught me, a young man of sixteen, to paste the following rule in my hat: In taking charge of a new plant always make it a point to examine the boilers, both inside and outside, as soon as possible.

ISAAC VIGGERS.

Everett, Wash.

Packing Hydraulic Elevator Pistons

The pistons in the four hydraulic elevators in the building where I am employed were packed a few days ago. The vertical cylinders are 15 feet long and 11½ inches in diameter and raise the car 60 feet. The water pressure is 40 pounds to the square inch. The valve is operated by a rack and pinion. The method of procedure was as follows: The bolts in the upper limit stop button were loosened and the button run up on the controlling cable 2 feet and again tightened. This allowed the car to be run up until the piston struck the bottom head of the cylinder. The car was then secured in this position (the piston having to be packed from the bottom) by means of a rope and tackle; one end being attached to the crosshead over the car, the other end to the sheave timbers. The pulling rope on the tackle was dropped to the lower flat and fastened there. This provided means for lowering the car by hand, if necessary.

It is the custom with many when packing this style of piston elevator to pass a strong rope around the car crosshead and sheave timbers, and depend on getting the cylinder full of water when ready to start. This is all right for those experienced in the work, but the use of the blocks and rope is practically no more trouble, and should a novice loosen the rope without having the upper part of the cylinder full of water, a wreck would surely result.

When the car was secured, the water-supply valve was closed, the upper cylinder air cock opened and the controlling valve thrown in the position for the car to ascend. The drain valve on the lower end of the cylinder was opened, and when the cylinder was empty the controlling valve was thrown in the position for the car to descend, which allowed the circulating pipe to empty. The bottom cylinder head was then removed, the bull head taken from the piston and the old packing pulled out and new packing put in. Instead of cutting the packing ⅛ inch short of a complete circle, the packing was cut ⅛ inch full and forced into place with soft-wood drivers. When ready, the cylinder was filled with water. To do this the drain valves were closed and the upper air cock left open, the controlling valve left in the position to descend and the water-supply valve slowly opened. When the cylinder was full, the upper air cock was closed and the controlling valve placed on the center. Then the tackle rope was untied and the slack carefully let out. The rope was then removed and the upper limit stop button put back in its original position. The car was then run up and down a few times, with the air cocks open to remove any entrapped air, and the job was done.

JAMES E. NOBLE.

Toronto, Can.

Crosshead Construction

In the style of crosshead shown in Fig. 1 the holes in the shoes are larger than the standard *A* and a knock will occur, which will gradually become worse, until the guides wear out of line, when a "see-sawing" motion will be set up. This will gradually become worse until the shoe is

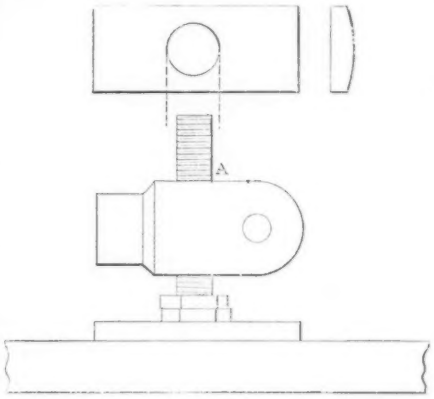


FIG. 1

slightly out of line, when turning and twisting of the piston, crosshead and connecting rod will occur at each stroke. This type of crosshead is never made with the pin in the center, and the heaviest thrust in running over comes in front of the shoe standard. Being out of balance, more weight is placed upon the front than elsewhere, thereby pressing the toe hard against the heel and raising it from the guide. A crosshead of this style is simply a makeshift.

The crossheads on a number of cheap engines are designed so that the wear is taken up on the top and not the bottom shoe, as it should be in all cases where the engine runs over. This is because the thrust comes there, which will eventually lower the piston rod and cause it to bind and scrape on the bottom part of the

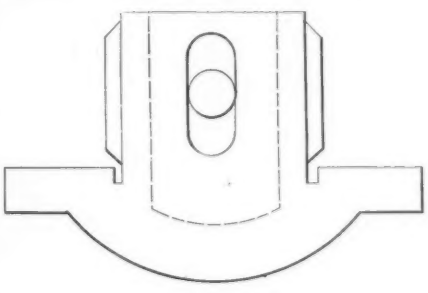


FIG. 2

stuffing box, causing the packing to wear quickly, and produce unnecessary friction and worry.

One engine builder makes the crosshead with an elongated hole for the piston rod, as in Fig. 2, and instead of screwing the rod up in the crosshead body properly and tightening up a jam nut, a nut is brought up against the body of the crosshead from both sides and tightened, leaving room for raising the rod in the slot-like hole when the shoe has worn enough to cause the rod to bind or travel out of line. This within itself is O. K., but to work out harmoniously in relation to other parts of the engine it is somewhat premature, as no provision is made for raising the crosshead pin. Consequently, when the shoe has worn any perceptible amount it brings the connecting rod just so much out of line with the piston rod, and as the thrust is downward on the guides and horizontal, or in the direction of reciprocation, in the shaft bearings, it is evident that the shaft will not lower itself in relation to the crosshead. After being in use long enough to have a considerable amount of adjusting done, wearing of the stuffing box is produced. A design that can be adjusted from the bottom as well as the top is much more preferable.

LLOYD V. BEETS.

Nashville, Tenn.

A Broken Dashpot Rod and a Peak Load

I was employed in a plant furnishing power for lighting and railway service, or, rather, what served as a substitute for service. We had a tandem compound-condensing Corliss engine which was the "whole show" on the peak load, and without it there was no possibility of pulling the load. One Saturday evening the head-end dashpot rod broke off close to the brass head. The valves had been set with the least possible lap, and the admission eccentric set back as far as possible in order to give the governor the greatest range of control, and the steam hook carried the block and the steam arm with it. As the lap was so small, the valve was not closing the port, because the shoulder of the hook did not throw the steam arm as low as the dashpot plunger had drawn it; therefore, the high-pressure piston was making the head-end stroke against this pressure of live steam, and the receiver pressure went up to about 70 pounds.

The entire lighting load, except about 30 amperes, was on this engine, and the lights grew a dull red. The oiler jumped for the throttle and was in the act of closing the valve when I came in from the boiler room and told him to wait until we got our bearings. I thought of two ways to operate the engine: I could lengthen the link rod to the head-end valve, giving it more lap, so that when the steam hook was carrying it the port would be closed, and in this way we could have used that end independent of the governor; or, we could leave the head-end valve closed and increase the cutoff on the crank end of the high-pressure cylinder, so as to raise the receiver pressure as high as possible. We did the latter, and the engine picked up the load, and

from that on until the load began to go off we had about 95 volts. This was not as good as 110, but it was better than none.

WILLIAM WESTERFIELD.

Lincoln, Neb.

Purification System a Paying Proposition

The water in Santa Clara valley contains scale-forming elements in abundance, especially bicarbonate and sulphate of lime, with some chloride of sodium. My predecessors had tried various kinds of boiler compound, but did not obtain satisfactory results, and if the compound acted on the scale-forming matter at all the scale was left in the boilers. At best the effect of the compound was slight and the boilers scaled badly.

We have three Heine boilers, rated at 125 horsepower each, which are worked to the limit. During my first year we had to renew thirty-three tubes on account of their leaking and crystallizing at the ends where they were expanded into the water leg. It also cost $100 per boiler each year to scale out the tubes and water drum. We have a good tube cleaner, but four to six tubes a day of ten hours was all we could clean. Added to this expense was the extra amount of oil fuel used to keep up steam, owing to the scaled condition of the boilers.

During the running season I opened up two boilers every Sunday, washed them out and cleaned them as well as I could in the limited time I had. During this time our firm installed a water-purifying plant, and last spring they put in a settling system. It cost complete about $900.

The system is simple, requires very little attention and is inexpensive to operate. I use 1½ pounds of caustic soda to each 1000 gallons of water. The soda is placed in a small tank, and 20 or 25 gallons of water added, then a little steam is turned on until the soda is dissolved in the hot water. Then the centrifugal pump is started and while the large tank is filling the soda solution is permitted to run in and mix with the water, so that when the water tank is full the soda tank is empty. I then let the water settle for 12 hours, if I can, as it is better to settle that long before using.

To test if the water has the proper amount of soda, use a solution of phenolphthalein, consisting of one part phenolphthalein to one part of grain alcohol. Use three drops in a glass of treated water and if the water turns very pink, use less soda; if too clear, use more soda. By a little experimenting one can tell the proper amount to use. It is also possible to tell by the amount of settling in the feed-water tank.

As a result of using this system, my boilers are clean, I have no burnt tubes to renew and the efficiency of the boilers has been increased 20 per cent. We expect the installation to pay for itself in two years.

B. F. COUSHMAN.

San José, Cal.

What Reversed the Polarity?

We have a three-wire direct-current system for light and power, 220 volts on the outside and 110 volts on the neutral wires. A part of our equipment consists of a turbine coupled directly to two 25-kilowatt 110-volt compound-wound generators, connected in the usual way in series. This unit is always run by itself and cannot be cut in with any of our other machines, running only when the load is within its capacity.

One Sunday at 8 a.m. the machine, which was running perfectly, was shut down and no other machine run until it was started at 11 a.m., when the voltmeter on one of its generators failed to register, but showed a tendency to work backward. Reversing the meter connection temporarily it registered properly, and on cutting the machine in our 110-volt circuit the lamps burned nicely, but we failed to get our 220 volts for motor purposes.

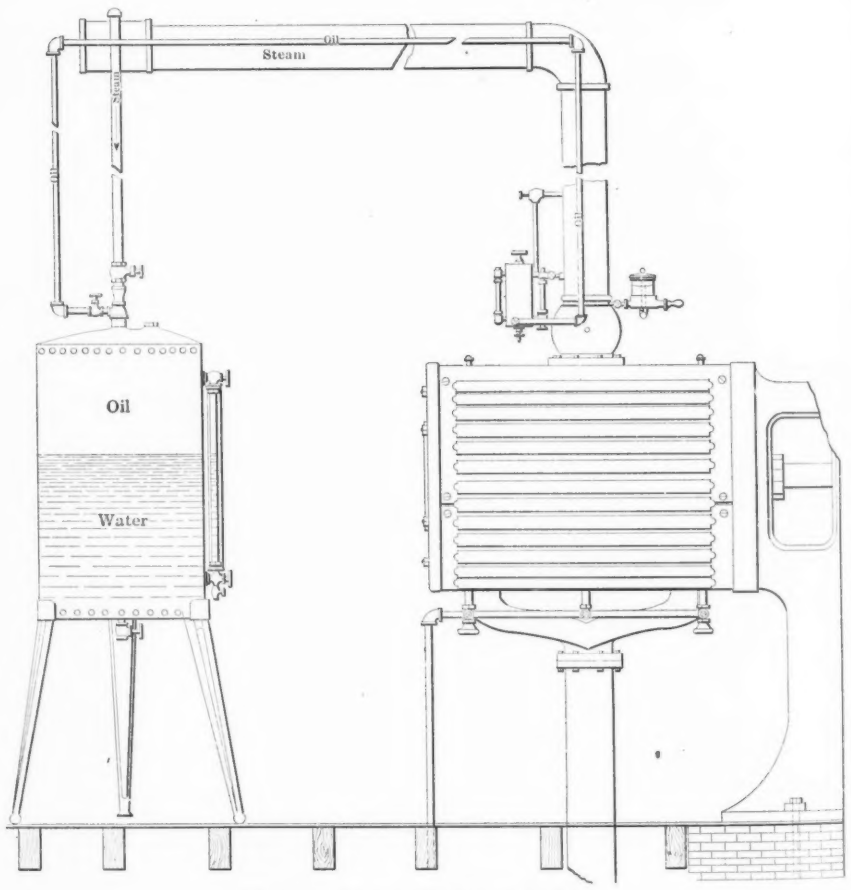
The indications were that something had reversed the polarity of one generator. The voltmeter having practically proven this, we reversed the field connections on the generator that was wrong, and got no current at all. Next we reversed the brush connections and everything was all right except that the voltage on the one generator would drop as the load increased. Next we reversed the compound-winding connections and everything was all right.

W. S. YOUNG.

Cherokee, Iowa.

A Self-filling Lubricator Attachment

The accompanying sketch shows a very satisfactory and economical device for supplying cylinder oil to a lubricator. The supply tank may be located in any convenient place in the engine room. An inspection of the diagram will be almost sufficient explanation.



SELF-FILLING LUBRICATOR ATTACHMENT

In operation, the water of condensation in the pipe tapping the steam main fills up the reservoir from the bottom, thus forcing the oil over into the lubricator. The device may be attached to any or as many lubricators as desired.

EDWARD T. BINNS.

Philadelphia, Penn.

A Heavy Engine Repair

In a power station of the Northern Texas Traction Company the crank disk on the high-pressure side of a 900-horse-power engine became loose, and it was decided to replace it with a new one, to be cast in a local shop. When the new disk was ready the work of removing the old one (weighing about 5000 pounds) was begun. This was accomplished by drilling holes through the disk along the diameter, as shown in Fig. 1. Taper pins were driven into these holes with a batt-

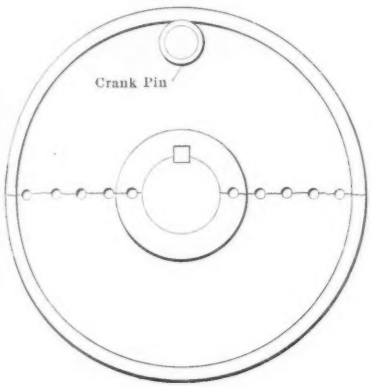


FIG. 1

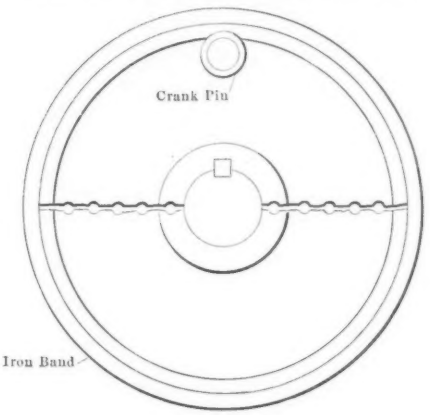


FIG. 2

EXHAUST-PIPE DRIPS

tering ram, which split the casting as shown.

When the new casting was shrunk on the shaft, owing to the hardness of the metal of which it was composed, it cracked from shaft to rim, by way of the crank pin, completely ruining the disk.

The engine was badly needed, and with the holiday rush coming on, something had to be done. Finally it was decided to use the old disk again until a new one of softer metal could be made. This was accomplished as follows: Two iron bands were forged from 3x3-inch iron bars, and bored to a diameter ⅛ inch less than the diameter of the disk. The edges of the two halves of the old disk were chipped away so they would not be drawn together by the bands, thereby insuring a tight fit on the shaft. The disk was also filed out slightly at the broken edges to prevent the sharp edges from nipping the shaft while being drawn together by the bands.

The two halves were put in place on the shaft and temporarily held by iron straps, bolted to the face of the disk across the crack. The iron bands were then shrunk around the disk, leaving the job as shown in Fig. 2, with the exception of a small iron band which was shrunk around the hub of the disk, not shown. This disk was run until a new one was made and from appearances would have lasted indefinitely.

The removal of the old disk when the new one was received was accomplished by putting a sheet-iron shield around the rim and playing the flame from an oil burner through a hole in the shield. This expanded the bands, allowing them to be removed.

C. L. GREER.

Handley, Tex.

Effect of Oil in Steam Boilers

At one time I was employed as engineer in a large manufacturing building which had two power plants. I had charge of the smaller. One day a new steamfitter advised the manager to have all the drains from all engines and pumps and the returns from the steam coils run into one tank and feed the boiler in the small plant from this tank by means of a pump. I advised the manager not to have the drains from the engines and pumps run into this tank, as I knew we would get oil in the boiler, due to the drains from the engines and pumps. No attention was paid to this advice, and after feeding the boiler from the tank about three weeks it was noticed that some of the tubes and seams in the shell leaked badly. As the boiler could not be shut down, the steam was blown out of the boiler at night and a boilermaker repaired the leaks. In a few days the boiler leaked badly again. It was opened up the next Sunday, when it was found to be in bad shape. The shell and tubes had a nice coating of cylinder oil. The boiler had to be shut down for a few days in order to remove the oil by thoroughly boiling the boiler out with soda ash and to repair the leaks. The drains from the engines and pumps were then removed from the tank.

Another case was that of a small plant in a shoe factory. This plant consisted of one return-tubular boiler, a slide-valve engine and a boiler-feed pump. The pump received its supply of water from an oil barrel located below the engine-room floor. This barrel was kept filled with water from the city main, the drain from the engine and pump also leading into this barrel. When it rained, the dirt from the yard would run into the barrel, there being no sewer in the engine room. There had always been trouble with the boiler leaking, due to cylinder oil coming with the feed water from the barrel. In order to remedy this trouble the barrel was done away with and a pipe was run direct from the water meter to the pump, while the drains from the pump and engine were run to a sewer.

H. A. JAHNKE.

Milwaukee, Wis.

Drip in the Exhaust Pipe

It is nothing uncommon to find exhaust pipes to either engine or steam pump connected as shown in Fig. 1, an arrangement which is very apt to give trouble, especially in cold weather.

If the exhaust pipe runs up, as it does in many places, from 30 to 50 feet, more or less condensation accumulates in the pipe, and must be carried to the top of the pipe, as there is no other escape, and thereby cause unnecessary back pressure.

A friend had a small engine with the exhaust piped this way. The throttle valve leaked a little, and one cold morning last winter he was unable to start his engine. The small amount of escaping steam had condensed in the exhaust pipe, where it froze. Had it been piped like Fig. 2 the water could have escaped through the drip pipe. It is a good idea to have the drip pipe covered in some way to prevent it from freezing in extreme cold weather. In some instances it is inclosed in a box packed with sawdust and run into a barrel sunk in the ground.

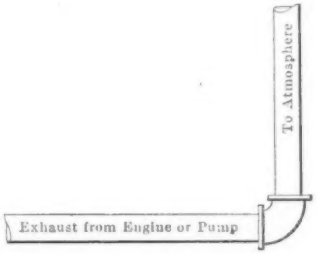


FIG. 1

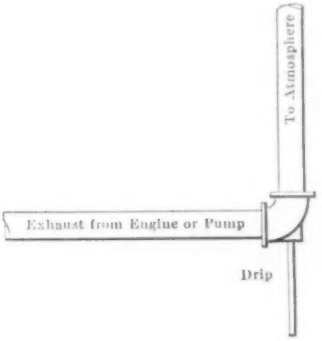


FIG. 2

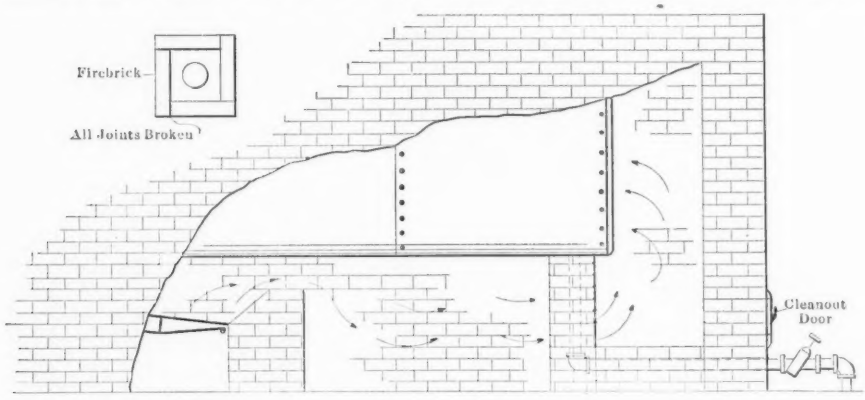
EXHAUST-PIPE DRIPS

F. P. KINDER.

Saxonville, Mass.

Protection for the Blowoff Pipe

The accompanying illustration shows the kind of blowoff-pipe protection I use in my plant. With this kind of protection none of the hot gases can strike the pipe,



SHOWING ARRANGEMENT FOR BLOWOFF PROTECTION

but when doing any repair work the brickwork is taken down, and in rebuilding it no mortar is used. In my 10 years' experience I have never had any burned blowoff pipes.

J. C. KAHL.

Blacklick, Ohio.

Wants Advice about His Commutator

I am running a 75-kilowatt 300-ampere 250-volt direct-current dynamo at a voltage of 220, operating at 750 revolutions per minute. I cannot keep my commutator from sparking badly at times. There are twelve carbon brushes.

We run about seven hours every night. After one night's run the commutator is bright in places, but is mostly dark, with streaks. Can some reader tell me what is the trouble?

A. L. BAKER.

Fisher, Ill.

Compounding Engines

In the early seventies, Cope & Maxwell, Cincinnati, O., brought out a line of compound duplex pumping engines, direct-acting without crank or flywheel, and I think quite a number of them were built. While structurally the two sides were combined as one machine, they were entirely independent as to operation, the only connection being the piping between the steam cylinders. I assisted at an economy test of one of them. The test gave a duty of about 60,000,000 foot-pounds per 100 pounds of coal. (Engineers had not learned to figure in heat units in those days.)

The most remarkable thing about that test was that about ten thousand dollars was spent in testing a six-thousand-dollar pump.

Some years after, I connected the exhaust of a 14x42-inch Corliss engine to the steam side of a 20x48-inch Corliss engine, both of which were operating geared pumps in the same building, but in separate rooms. As far as I know, no test was ever made of the economy of this plant, but the operation was satisfactory.

A few years ago it began to be noised around among blast-furnace operators that compounding was a good thing, and as the desire for increased production from the furnace led to increasing the blowing-engine capacity, it was quite the fashion to install another blowing engine for the plant, with a steam cylinder proportioned to act as the low-pressure side for the existing engine; the new engine using the exhaust steam from the old one, but otherwise being entirely independent, so the two engines might, and usually did, run at different speeds.

I have in mind a case where I worked on the drawings of a blowing engine with an 80-inch steam cylinder and an 84-inch air cylinder, both having a 60-inch stroke. It was erected alongside another blowing engine having a 42-inch steam cylinder and a 60-inch stroke. The new engine used the exhaust from the old one, and exhausted into a barometric condenser, practically doubling the air capacity for the furnace, with very little increase of fuel.

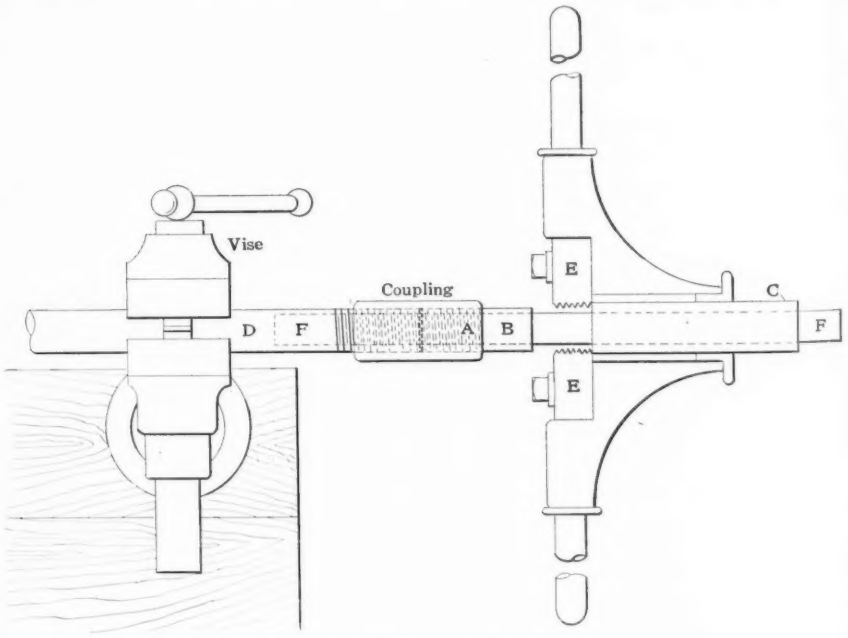
ALONZO G. COLLINS.

Philadelphia, Penn.

How to Thread Close Nipples

Close nipples are difficult to cut, in the usual manner, without injuring the threads. The following is a method for preventing this trouble: Cut a thread *A* (see sketch) on the end of a pipe of the size of nipple required and cut the pipe the right length for the nipple. Next cut a long thread on a piece of pipe *D* so it will easily screw about three-fourths through the coupling. Then screw the threaded nipple *B* into the coupling until its end rests against the end of the piece *D*. The coupling is then screwed tight on *D* with a wrench and the nipple *B* will be held firmly and can be easily threaded.

As the coupling will prevent cutting a very close nipple in the usual manner, reverse the stock and dies as shown at *E E*, and the pipe *B* can then be threaded close up to the coupling. To guide the dies and stock when used in this manner, get an iron rod, or a strong wooden rod *F*, making a snug fit inside of nipple to be cut and long enough to extend about 6 inches in through the nipple, and out through stock. Then slip the stock over this rod and down on the pipe in position



HOW TO THREAD A CLOSE NIPPLE

ready for cutting. Then put a piece of pipe *C* of the same size of the nipple *B* over the rod *F*, in through guides of the stock, resting against the back of the dies. This will guide the dies so that they will cut the thread on the nipple parallel to the thread already cut.

F. E. FICK.

Govans, Md.

Pressure on Both Sides of a Valve Disk

Several years ago I read an article that stated that the pressure on the back of a valve was equal only to the unbalanced part of the valve or port opening, multiplied by the steam pressure in the steam chest. In the October 27 issue, page 702, Mr. Anderson advances practically the same idea. As a great many have to be "shown," myself included, I constructed the apparatus shown in the accompanying sketch, to prove more positively to myself and others the correctness of the idea.

The lengths and sizes of the pipe and fittings shown in this sketch have nothing to do with the principles involved; I used them because they were at hand, and any other sizes or lengths would do as well. The vertical check valve is cut away to show the disk and seat. I made a new disk somewhat larger in diameter than the original, in order to get a greater difference in areas of top and bottom of the valve disk.

The sizes and lengths of the pipe shown are as follows: The large pipe is 2 inches in diameter and measures 5 feet 8 inches from the valve seat to the top. The smaller pipe is ⅜ inch in length and extends above the valve seat 5 feet 9 inches. The 2-inch pipe was clamped in a vise in a vertical position and filled with water, obtaining a pressure of

$$5.66 \times 0.434 = 2.4566$$

pounds per square inch on the upper side of the check valve.

To use Mr. Anderson's method of figuring to obtain the pressure necessary to produce equilibrium in the check valve, the area of the upper side of the check valve, which was 2.5625 inches in diameter, is 5.15 square inches; the area of the under side of the check valve exposed to the opening of the valve seat is 2.0625 square inches, the diameter being 3.35 square inches. Therefore

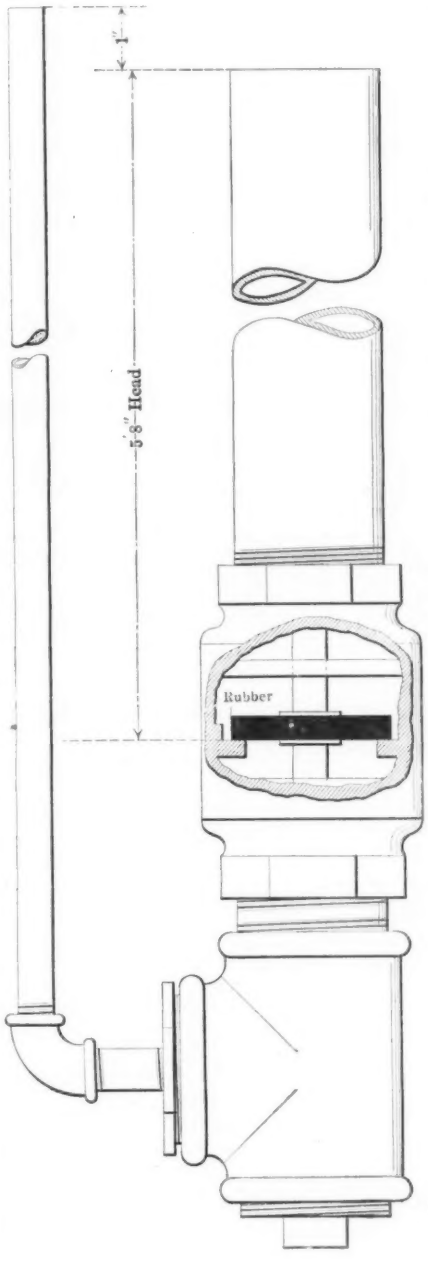
$$\frac{2.4566 \times 5.15}{3.35} = 3.77$$

pounds, the pressure required on the under side to place the check in equilibrium, or equal to a head of water in small pipe of

$$\frac{3.77}{0.434} = 8.68$$

feet above the valve seat, or 3 feet 1.36 inches above the top of the 2-inch pipe. So much for figures and theories; now for practice:

After the 2-inch pipe had been filled with water, the small pipe was very slowly filled to prevent the force of the falling water acting on the under side of the valve. The water raised in the small pipe to within ¾ inch of the top, or ¼ inch above the top of the 2-inch pipe before the 2-inch pipe began to overflow. All the water was then let out, the valve



MR. LIVINGSTON'S APPARATUS

seat and disk coated with white lead and oil and replaced, and the operation repeated. Water raised in the small pipe to within ¼ inch of the top before the large pipe began to overflow. A small stream of water about the size of a pencil point was kept running in the small pipe for some time, with probably no more than ¼ inch variation in the head in the small pipe, and the large pipe overflowed continuously.

R. S. LIVINGSTON.

Deweyville, Tex.

An Interesting Repair on a Cylinder

I had erected a large, horizontal pumping engine with a steam-jacketed Corliss steam cylinder, about 30 inches in diameter, and was doing the preliminary running and testing prior to turning it over to the purchaser.

I had been puzzled for some time for the cause of the high expansion curve of the indicator diagram from the head end of the cylinder, and attributing it to a leaky steam valve. I used up a lot of (as it afterward turned out, misdirected) energy in scraping that valve to a fit.

One day, for some reason that I have now forgotten, it was necessary to disconnect the steam-jacket pipe from the cylinder, but I did not shut down the pump, letting it run without steam in the jacket. When the pipe was opened, I was surprised to see a puff of steam escape from it every time the engine took steam on the head end, and concluded that a hole that would pass steam from the cylinder into the jacket space would also let it go the other way; it was a very natural surmise that steam from the jacket was the cause of the abnormal diagram.

As soon as opportunity offered to shut down the pump I took off the cylinder head and, turning steam into the jacket, was almost deluged with the hot water and steam that poured out of innumerable pin-holes around about two-thirds of the circumference of the counterbore of the cylinder, caused either by dirt in the iron, or dirt that had washed from the sand of the mold in casting.

I had often closed small holes in castings by peening around them with a round-peened hammer, but had never tackled so extensive a surface as this. I concluded to try it, however, as I could only make a bad matter worse, and the cylinder was useless as it was. Besides, as all the leaks were in the counterbore, the peening would not injure the piston fit in the slightest.

I succeeded in making it tight, but it took an enormous amount of hammering, and was a hot, disagreeable job, as it was necessary to keep pressure in the jacket so as to locate the leaks.

I suppose the reason the defect was not discovered while erecting the engine was that the slush that had been put on in the shop to prevent rust was not as thoroughly cleaned off in the counterbore as in the parts that were swept by the piston.

ALONZO G. COLLINS.

Philadelphia, Penn.

Exhaustion of Ignition Batteries

In answer to the inquiry in the November 10 number, under the above heading, I would say that a set of dry batteries will last nearly as long with the engine running at full speed as half speed, as the duration of contact will be only one-half as long, while a set of wet batteries of the Fuller type would last probably one-half longer with the increase of speed and the shorter time of contact.

HENRY B. HICKS.

Oriskany Falls, N. Y.

Amount of Water Evaporated per Pound of Coal

Following is the result of a test relative to the amount of coal used to evaporate a certain number of pounds of water: During the test we evaporated 1,332,500 pounds of water from a feed-water temperature of 180 degrees into steam at 100 pounds boiler pressure, and used 264,000 pounds of coal, or

$$\frac{1,332,500}{264,000} = 5.47$$

pounds of water from 1 pound of coal.

The steam table gives the number of heat units required to raise 1 pound of water from 180 degrees into steam at 100 pounds pressure as 1036.9 B.t.u. Therefore, from 1 pound of coal we obtained

$$1036.9 \times 5.47 = 5671.84$$

B.t.u. The actual results, therefore, fell far below the standard of 14,500 B.t.u. for 1 pound of coal.

E. E. EDWARDS.

Des Moines, Iowa.

Crack in a Lap Seam Boiler

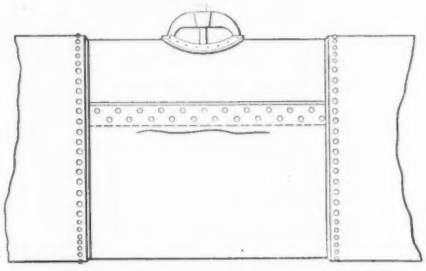
The editorial in the December 15 number, relative to the discovery of a crack in a lap-seam boiler at the Harwood Counter Company's factory, and the prevention of an explosion by the intelligent and prompt action of the engineer, who shut down the plant, prompts me to contribute the following account of how it happened:

On Saturday, November 28, the plant had been shut down for the week and the engineer had gone to the top of the boiler to close the stop valve, when he noticed steam issuing from the brickwork over the boiler. Upon removing some of the bricks he found that the steam was coming from a crack 18 inches in length (see sketch) along the longitudinal joint of the middle sheet.

On Sunday, hydrostatic pressure was applied and resulted in the crack extending nearly the length of the seam by the time 100 pounds pressure was reached.

The boiler was installed 19 years ago. It is 60 inches in diameter and 16 feet long, and consists of three sheets. The longitudinal joint is of the double-riveted lap-seam type. The crack was not located under the lap, as has generally been found to be the case, but ran parallel to the edge of the overlapping plate. The boiler was insured and was inspected last April, a pressure of 75 pounds being allowed.

It would seem that the repeated failures of boilers having this type of joint should be sufficient demonstration that the laws governing the use of them are not all that they should be, and that something further than prohibiting their manufacture is necessary, if the public is to be protected. The kind of logic which condemns a boiler before it is built and yet allows boilers of the same construction to remain in use after five to forty years of service is typical of Massachusetts. Perhaps if a law were passed compelling the owners of these lap-seam boilers to post in conspicuous places about the building notices reading, "Dangerous Lap-seam



WHERE THE CRACK OCCURRED

Boilers are Installed in This Building," there would be less need of legislation along this line.

ARTHUR F. CLAWSON.

Lynn, Mass.

The Modern Surface Condenser

Referring to Otto H. Mueller's criticism of the surface-condenser article under my signature in the August 11 number, would say that I am pleased that the article in question brought at least one answer. It is unfortunate that I have not had access to Professor Josse's paper and, therefore, could not discuss his results, but Mr. Mueller's information concerning the vertical-tube condenser seems to be much different from that which has come to this country.

I note, however, that he agrees with me regarding the main thesis that in ordinary condensers only a part of the surface can do work and that improvements can be made by avoiding "idle corners and pockets." I fail to understand how surface efficiency can be obtained by excessively large air pumps or the vacuum augmenter. Where good surface efficiency is possible and there are no serious air leaks, the air pump of ordinary size is usually more than sufficient for the work, but without good surface efficiency any amount of air pump usually fails to maintain a good vacuum.

His statement that in the best modern condensers with very large air pumps, it is impossible to get a condensation of more than 12 pounds per square foot seems rather curious in the face of many test results which have been obtained in an experimental way. The condensation of 18 or 20 pounds is certainly possible, and we have reports from careful experimenters who have gone as high, under certain conditions, as 40 or 50 pounds per square foot of surface. These figures are not commercial, but serve to show that surface efficiency can be obtained under proper conditions.

His remarks regarding Weighton's test are somewhat misleading. It is true that his circulating water was cool, and in most of the tests runs from 40 to 52 degrees, but his vacua were rather uniformly high and the condensation per square foot of surface was more than remarkable, in the light of Mr. Mueller's statement that 12 pounds is the practicable limit of condensing. In looking over Weighton's tests I note condensations as high as 35 pounds per square foot per hour with a 28-inch vacuum and 41 degrees condensing water. That the high rates of condensation are due to the cool circulating water is, of course, a fact, but he will find similar results with warmer water where the circulating pump is sufficiently large to keep the average difference between circulating and condensing water as great as possible.

Referring to the rate of heat transmission being proportional to the cube root of the velocity of the water in the tubes, I refer Mr. Mueller to the experiments of Professors Ser and Joule, whom he himself quotes, and also to Hausbrand's book on "Evaporating, Condensing and Cooling Apparatus," with which he is probably familiar.

Referring to the diagram of heat transference, Mr. Mueller will note that I said the diagram showed what might be expected in the way of heat transference under various conditions. This diagram represents all that I have been able to find in an experimental way on the subject of heat transference through tubes, and I should be very glad to modify this diagram with the results of further and more accurate experiments whenever they are made public. It would be of value to the profession if Mr. Mueller would put the work of Professor Josse before the readers of POWER, as publications of the character in which his work appeared are not usually available in this country.

In conclusion, I may say that if I have been successful in stirring up manufacturers to a point where they will attempt to design their condensers on a more scientific basis and endeavor so to arrange the surface that there will be no "idle corners and pockets," the article will have served its purpose.

G. A. ORROK

New York City.

Condensers and the Economy of Their Use

Usual Types, Their Action, and Economy of the Condenser as Compared to a Noncondensing Plant Using Exhaust Steam for Heating

BY CHARLES L. HUBBARD

The saving which may be obtained by the use of a condenser in connection with a steam power plant depends upon several conditions, and must be carefully worked out in each particular case. One of the most important is the cost of condensing water, which, if purchased at city rates, may call for an expenditure far in excess of the saving in coal. Other important factors are the size and water-rate of the engine and the disposal of exhaust steam.

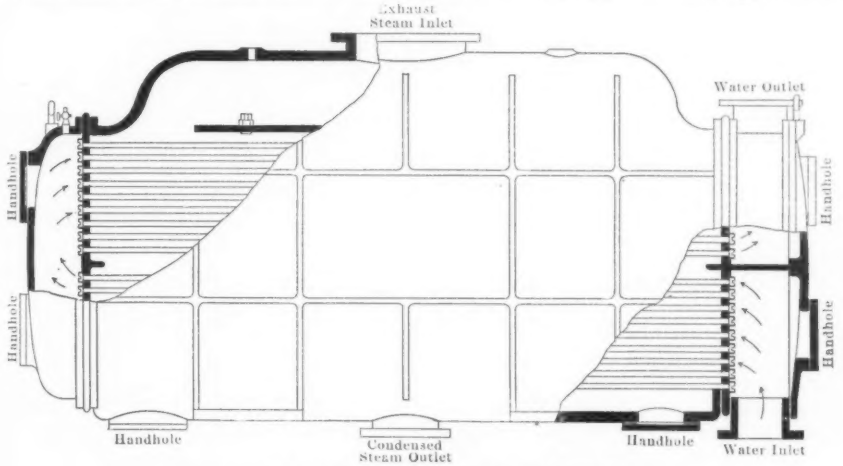


FIG. 1. SINGLE-TUBE SURFACE CONDENSER

Before taking up a comparison of the relative costs of operating a plant under different conditions, both condensing and noncondensing, it may be well to consider some of the underlying principles upon which a condenser operates and also describe some of the types in common use.

As the pressure of the steam is reduced in the cylinder of an engine, due to its expansion, it soon reaches a point where it is not sufficient to do work economically, and must, therefore, be exhausted either into the atmosphere or into a chamber at some pressure below it. Under the first condition, which corresponds to a noncondensing engine, it is evident that the steam cannot be expanded below atmospheric pressure and also that the piston must work against a pressure of 14.7 pounds on the return stroke. If this back pressure can be reduced, or better, entirely removed, there is, of course, a substantial gain in the power of the engine, and also the steam can be expanded to a lower pressure and thus change more of its heat into useful work.

These results are obtained in practice to a greater or less degree by the use of a condenser, which consists of a closed chamber into which the steam is exhausted from the engine. Practically instantaneous condensation is produced by the application of cold water which takes up its heat. As the process of condensation from steam to water reduces the volume to about 1/1600 of what it was originally, it is plain that a large part of the closed chamber is empty and, therefore, under practically no pressure, or in a state of vacuum. As this condensation occurs immediately after exhaust takes place, the same condition holds for each successive stroke of the engine.

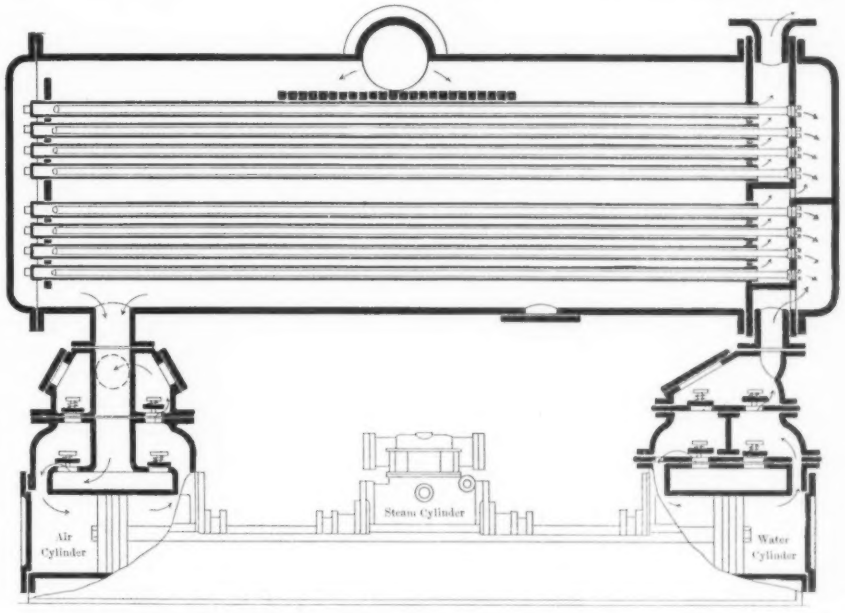


FIG. 2. DOUBLE-TUBE SURFACE CONDENSER AND COMBINED CIRCULATING AND AIR PUMP

Condensers are of two general forms, known as surface condensers and jet condensers. In the first of these the cool condensing water is passed through a series of thin copper or brass tubes and the condensation is caused by the steam coming in contact with their outer surfaces. In the case of the jet condenser the steam and water come in direct contact, the latter being introduced into the condensing chamber in the form of a spray. Although in theory a perfect vacuum should be obtained in this manner, it is never realized in practice. This is due partly to the water being heated to a certain extent so that condensation is not complete, partly to the in-leakage of air through joints and stuffing boxes, and in the case of jet condensers a certain amount of air is drawn in with the cooling water, which tends to destroy the vacuum.

The action of a condenser is made continuous by providing a constant supply of cooling water, which is furnished by means of pumps, one for circulating the cooling water through the pipes, and another for removing the condensed steam and air from the condensing chamber. A jet condenser requires only one pump, the condensed steam and cooling water being mixed, and removed together.

The Surface Condenser

Fig. 1 shows the general form and con-

struction of a surface condenser. The condensing chamber is seen to be filled with tubes through which the cooling water is caused to circulate, as shown by the arrows. The exhaust steam enters the top and is spread by means of a baffle plate, placed just below the inlet in order to avoid cutting the tubes. The condensed steam and air are drawn off from the bottom by the air pump. Fig. 2 shows a combined circulating and air pump, a form commonly used in connection with surface condensers. The steam cylinder is placed at the center of a long piston rod, with the water piston at one end and the air-pump piston at the other.

The arrangement of the tubes varies in different designs and makes. Fig. 1 shows the single-tube pattern while Fig. 2 illustrates a form of double tube. This is shown in detail in Fig. 3. Condenser tubes are commonly made ¾ inch in diameter and of solid drawn brass tinned upon both sides. In order to provide for the unequal expansion of the tubes and shell, it is necessary in the case of single tubes to allow for independent expansion. This is done by means of screwed glands and stuffing boxes packed with cotton cord or a specially prepared tape, as shown in Fig. 4. In some cases the tubes are made as small as ½ inch in diameter, while in others they may be made as large as an inch, but ¾ inch is the more common size.

The tube sheets or plates to which the ends of the tubes are attached are of brass, and commonly 1.1 to 1.5 times the diameter of the tubes in thickness. The way in which the connections are made has an important bearing upon the thickness. With glands and tape packing, a thinner plate may be used than when the packing extends entirely through it. The tubes are commonly arranged in a zigzag manner, pitched from 1.5 to 1.7 of their diameter on centers. Tubes, plates, ferrules, nuts and washers should all be of brass in order to prevent corrosion. The shell is commonly made of cast iron; no wrought iron or steel should be used when the parts are exposed to the distilled water.

Computing the Cooling Surface

Under any given condition, the cooling

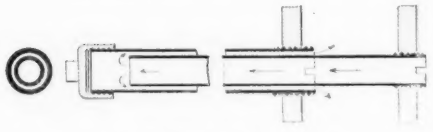


FIG. 3. DETAIL OF DOUBLE TUBE

surface required may be found from the following formula:

$$S = \frac{W\,L}{180\,(T - t)},$$

in which

S = Required cooling surface in square feet for brass tubes,

W = Weight of steam to be condensed per hour,

T = Temperature of the steam at condenser pressure,

t = Average temperature of the circulating water and

L = Latent heat of the steam at temperature T.

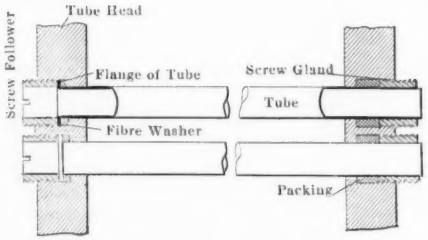


FIG. 4. ALLOWING FOR EXPANSION OF SINGLE TUBE

The use of this formula is best illustrated by applying it to a practical example. Take the case of a 300-horsepower engine using 22 pounds of steam per horsepower-hour, and assume the probable pressure in the condenser under average conditions to be 3 pounds. The average initial temperature of circulating water may be taken at 70 degrees in summer, and its final temperature, after passing through the condenser at 120 degrees. From this data we have:

$W = 300 \times 22 = 6600$ pounds of steam per hour.

$T = 141$ degrees, which is the temperature of steam at 3 pounds pressure, taken from a steam table.

$$t = \frac{70 + 120}{2} = 95$$

degrees.

$L = 1015$, latent heat of steam at 3 pounds pressure, from steam table.

Then substituting in the formula:

$$S = \frac{6600 \times 1015}{180\,(141 - 95)} = 800 +$$

square feet, or

$$\frac{800}{300} = 2.7$$

square feet per horsepower.

In marine practice it is common to make the cooling surface from 1.1 to 1.5 square feet per horsepower for the best types of triple-expansion engines. This figure is lower than the value just obtained, owing to the low steam consumption of this class of engine. A rule in more or less common use is to make the cooling surface one-half that of the heating surface of the boiler supplying the engine. This, however, gives a much larger surface than that obtained by the use of the formula. Thus, in the example just taken, suppose the boiler to be of the water-tube type, rated at 10 square feet of heating surface per horsepower. The total weight of steam used by the engine was 6600 pounds per hour; therefore, the rated horsepower of the boiler required to supply this amount of steam will be

$$\frac{6600}{34.5} = 191 +$$

and

$$191 \times 10 = 1910$$

square feet of heating surface in the boiler. Then, by the rule of thumb previously given, 955 square feet of cooling surface would be required in the condenser, or

$$\frac{955}{300} = 3.18$$

square feet per horsepower, as against 2.7 by the more exact method.

Surface condensers are more bulky than the jet condenser for a given capacity, which gives the latter a decided advantage in some cases. For marine work the surface condenser is used almost exclusively, as the sea water may be used for cooling, the condensed steam being used over and over again in the boilers. This calls for special care in the removal of oil, and also a certain amount of fresh water

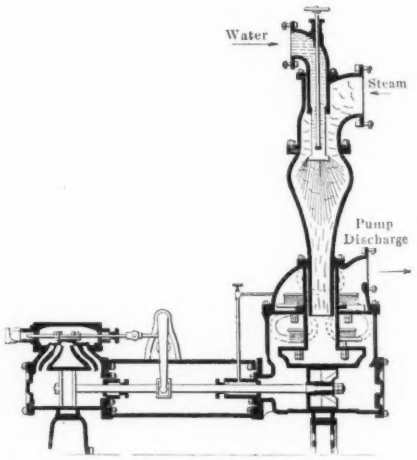


FIG. 5. A COMMON FORM OF JET CONDENSER

must be added at regular intervals. This should amount to 10 or 12 per cent. of the total weight of water fed into the boilers in a given time.

Jet Condensers

Fig. 5 shows a common form of jet condenser. This requires a smaller quantity of water at a given temperature to condense a given amount of steam than the surface condenser, because practically the whole of the cooling water comes in contact with the steam. If the boiler feed is taken from the discharge from the condenser, the cooling water must of course be from a source which makes it suitable for this purpose.

Referring again to Fig. 5, the cooling or injection water is supplied at the top through an adjustable spray cone which breaks it into small particles and thoroughly mixes it with the inflowing steam, and thus produces rapid condensation. The adjustable action of the cone is evident from an inspection of the cut. The mixture of condensed steam and

cooling water are drawn from the bottom of the condensing chamber into the air pump and delivered either to the sewer or hot well, depending upon whether the discharge water is to be used for feeding the boilers. If this is done, the exhaust steam should be passed through an efficient grease extractor before reaching the condenser. In designing a condenser of this type, it is customary to

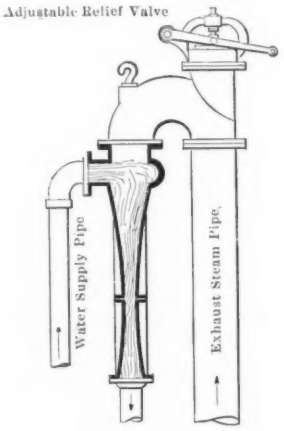


FIG. 6. THE SPIROJECTOR

make the volume of the condensing chamber from one-third to one-half that of the engine cylinder with which it is to connect. The method of delivering the cooling water to the condenser will depend upon circumstances; if the elevation is not over 20 feet above the source of supply, it may be drawn up by suction, but for higher elevations it is necessary to use a pump.

The "spirojector," so-called, shown in Fig. 6, is another type of the jet condenser. The peculiar construction of the inlet gives the water a rotary motion which produces a strong suction, thus drawing in the steam and air and rapidly condensing the former. The method of supplying the injection water is the same as in other jet condensers, and depends upon the elevation of its parts above the source of supply.

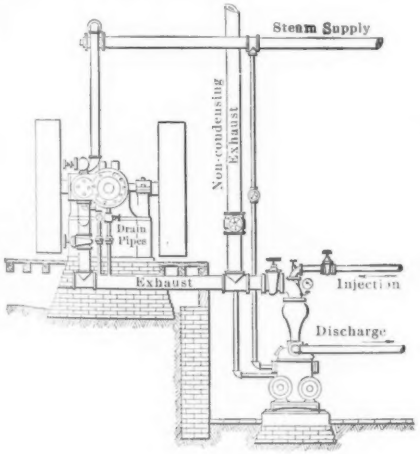


FIG. 8. A JET CONDENSER INSTALLATION

In Fig. 7 is shown an apparatus embodying some of the features of both the surface and jet condensers. This is made up of a steel shell filled with a large number of small brass tubes open at the ends to the atmosphere. The exhaust steam enters this shell and surrounds the tubes, and the condensation and air leakage are removed by an air pump in the usual manner. The general arrangement of the different parts is clearly indicated in the cut. Steam enters as shown, and at one end where the tubes are exposed, is arranged a fine spray of water, which, as it strikes the ends of the warm tubes, is vaporized and drawn through them by the action of an exhaust fan connected with the other end. The evaporation of the spray absorbs a large amount of heat from the tubes as it passes through them, and thus causes a rapid condensation of steam within the chamber. The pipe connections for a jet condenser will vary with different locations. Figs. 8 and 9 show common methods which may be varied to suit local conditions.

TEMPERATURE AND QUANTITY OF COOLING WATER

In connecting up any type of condensing apparatus, provision should be made for exhausting either through the condenser or outboard as may be desired. The elevation of the condenser with reference to the engine does not matter so long as the pipes are properly dripped. It is common practice in large power plants to place the condensing apparatus beneath the engine-room floor.

The gain in power or saving in fuel by the use of a condenser depends principally upon the steam pressure and the cutoff of the engine and also upon the temperature of the cooling water.

The quantity of water necessary to condense the exhaust steam from an engine depends upon the temperature of the water and the pressure and weight of the steam. Assuming average conditions, say with 26 inches of vacuum and cooling water at an average of 70 degrees, the required weight of water will be from twenty to thirty times that of the steam condensed. The lower the temperature of the water, the less the quantity required, and as the temperature increases, more is required. For example, at 80 degrees the weight of cooling water is thirty-five times that of the steam condensed, and at 90 degrees fifty-two times. The amount in each special case can be computed theoretically by the following equation:

$$Q = \frac{S - (t\,d - 32)}{t\,d - t\,e},$$

in which

Q = Weight of injection water per pound of steam,

S = Total heat (above 32 degrees) of a pound of exhaust steam.

te = Temperature of injection water and

td = Temperature of discharge water.

In using this formula it is necessary to know the terminal pressure of the steam in the cylinder at the end of expansion, in order to determine the total heat of the exhaust steam. This can be determined near enough for all practical purposes from Table 1.

The temperatures of the injection and discharge water must be assumed. In practice it is customary to take the former

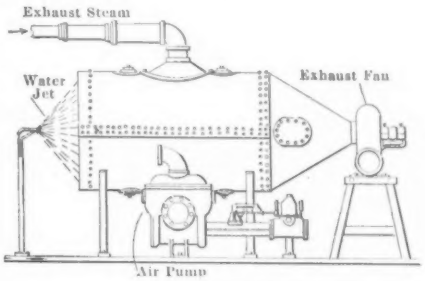


FIG. 7. A COMBINATION SURFACE AND JET CONDENSER

at about 70 degrees as a fair average in summer time. The temperature of the discharge water generally ranges from 110 to 120 degrees for ordinary conditions of vacuum, which means a vacuum of from 24 to 26 inches. Probably 70 degrees and 110 degrees represent about the average, and these may be taken in making the computations for the quantity of condensing water required. In making computations for a year's run, one should take the average temperature of injection water at about 50 degrees.

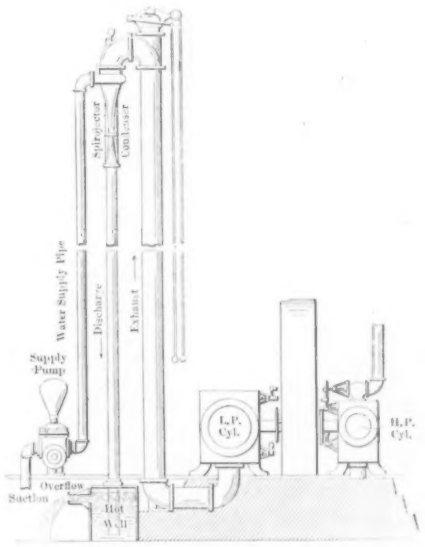


FIG. 9. THE SPIROJECTOR IN SERVICE

The following examples and illustrations, while revised and practically new, follow along the same general lines as those given by the writer some years ago in other publications, notably in the *American Electrician*. We trust, however, that this will not detract from their present value.

Assume the case of a 300-horsepower engine with 3 per cent. clearance, using

22 pounds of steam per horsepower-hour, and cutting off at 1/6 stroke; a boiler pressure of 120 pounds gage; and temperatures of injection and discharge water, 80 and 120 per cent., respectively. Find the quantity of condensing water required per hour.

Table 1 shows that the terminal pressure ratio is 0.191, and as the initial pressure is 120 + 15, the terminal pressure is

TABLE 1.
TERMINAL PRESSURE RATIOS.

	Per Cent. Clearance.					
Cutoff.	0	1.75	3	5	7	9
1–10	0.1	0.115	0.126	0.143	0.159	0.174
1–8	0.125	0.14	0.150	0.167	0.182	0.197
1–6	0.167	0.181	0.191	0.206	0.221	0.235
1–5	0.2	0.214	0.223	0.238	0.252	0.266
1–4	0.25	0.263	0.272	0.286	0.299	0.312
5–16	0.312	0.324	0.332	0.345	0.357	0.369
1–3	0.333	0.344	0.353	0.365	0.377	0.388
3–8	0.375	0.386	0.392	0.405	0.416	0.427
2–5	0.4	0.410	0.417	0.428	0.439	0.449
7–16	0.437	0.447	0.454	0.464	0.474	0.484
1–2	0.5	0.509	0.514	0.524	0.533	0.541
5–8	0.625	0.632	0.636	0.643	0.649	0.656
3–4	0.75	0.754	0.755	0.762	0.766	0.771
4–5	0.8	0.804	0.806	0.809	0.813	0.816
9–10	0.9	0.902	0.903	0.905	0.906	0.908

To find the (theoretical) absolute terminal pressure, multiply the absolute initial pressure by the number opposite the given cutoff and in the column corresponding to the percentage of clearance.

$$(120 + 15) \times 0.191 = 25 +$$

pounds, and a steam table shows that the corresponding total heat is 1155. The following values, therefore, may be substituted in the formula: $S = 1155$, $te = 80$ and $td = 120$. So that

$$Q = \frac{1155 - (120 - 32)}{120 - 80} = 26.7$$

pounds.

The total weight of steam used per hour by the engine is

$$22 \times 300 = 6600.$$

Therefore,

$$6600 \times 26.7 = 176{,}220$$

pounds of condensing water required per hour. This is equal to

$$\frac{176{,}220}{8.3} = 21{,}230$$

gallons, approximately.

The reason for the above formula is explained as follows: One pound of steam at 25 pounds pressure (absolute) contains 1155 heat units above 32 degrees; that is, if the steam were condensed and cooled to 32 degrees, 1155 heat units would be given off. But in the example it is cooled only to 120 degrees, and as one heat unit will raise the temperature of one pound of water one degree, one heat unit will remain in the condensed steam for each degree that its final temperature remains above 32 degrees. Therefore, the heat given out by condensing one pound of steam at 25 pounds pressure and cooling it to 120 degrees is

$$1155 - (120 - 32) = 1067$$

heat units. The heat absorbed by one pound of water in raising it from a temperature of 80 to 120 degrees is

$$120 - 80 = 40$$

heat units. Therefore,

$$\frac{1067}{40} = 26.7,$$

the pounds of water which will be required to absorb the heat given off by the condensation and cooling of one pound of steam under the conditions stated in the example. The matter of subtracting 32 degrees from temperature *td*, affects the result but little, and is often omitted in practice, and the formula written:

$$Q = \frac{S - td}{td - te},$$

but for purposes of demonstration it is necessary to include it.

The Gain in Condensing

The gain to an engine by condensing the exhaust steam acts in two ways, as already stated; it may cause a saving in steam consumption when delivering the same power, or the power may be increased with the same steam consumption. The force exerted by the piston of an engine during a single stroke is directly proportional to the difference in the pressures acting upon its steam and exhaust sides; and any reduction in pressure upon the latter adds a like amount to the steam side, and is therefore a net gain in power. Assuming that a vacuum of 26 inches of mercury may be obtained upon the exhaust side of the piston by the use of a condenser and making allowances for the cost of production, we may consider the gain as equivalent to about 12 pounds per square inch in the mean effective pressure. This is made evident by an inspection of Fig. 10, in which the full line represents the diagram with the engine running noncondensing. Now by using a condenser and keeping the points of cutoff and release the same as before, the back-pressure line will be brought down to the position shown as dotted.

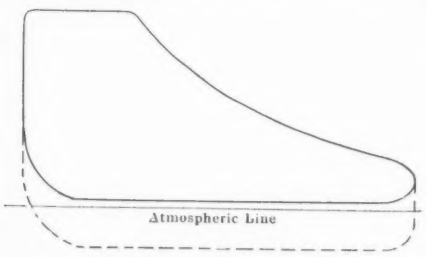


FIG. 10. AREA ADDED TO DIAGRAM BY CONDENSER

The hight included between the original and final back-pressure lines represents the gain in pressure, and as it lasts throughout the stroke, it may be added directly to the mean effective pressure.

This gain under ordinary conditions amounts to about 12 pounds, as already stated. The gain in power, therefore, by running an engine condensing bears the same relation to the power developed when running noncondensing as 12 pounds to the mean effective pressure when noncondensing. Written in the form of an equation, we have

$$\frac{p}{P} = \frac{12}{M.E.P.},$$

in which

p = Gain in power,
P = Power when running noncondensing,
$M.E.P.$ = Mean effective pressure when running noncondensing.

If this is written in the form

$$p = \frac{12\,P}{M.E.P.},$$

it gives an easy way of finding the probable increase by condensing, when the power and mean effective pressure are known.

Approximate Mean Effective Pressure

The approximate mean effective pressure for any given case may be found by the use of Table 2. The ratios given in this table are for absolute pressures, and their use can best be shown by a practical example. A noncondensing engine with 3 per cent. clearance cuts off at 1/5 stroke; the boiler pressure is 80 pounds gage, or 95 pounds absolute. What is the mean effective pressure? The ratio for 1/5 cutoff is 0.545; therefore, the mean effective pressure is

$$95 \times 0.545 = 52$$

pounds, but this is on the assumption that the engine exhausts into a perfect vacuum.

Under ordinary working conditions the back pressure acting on the piston of an engine exhausting into the atmosphere will be about 2 pounds above atmospheric pressure, or 17 pounds absolute, so that in the above example we must subtract 17 from the result found, which will give

$$52 - 17 = 35.$$

TABLE 2.
MEAN PRESSURE RATIOS.

	Per Cent. Clearance.					
Cutoff.	0	1.75	3	5	7	9
1–10	0.330	0.354	0.369	0.392	0.413	0.432
1–8	0.384	0.405	0.418	0.438	0.457	0.474
1–6	0.465	0.481	0.492	0.509	0.524	0.538
1–5	0.522	0.535	0.545	0.559	0.572	0.584
1–4	0.597	0.607	0.615	0.626	0.636	0.646
5–16	0.676	0.684	0.689	0.698	0.706	0.713
1–3	0.699	0.707	0.713	0.719	0.727	0.734
3–8	0.742	0.749	0.753	0.759	0.765	0.771
2–5	0.766	0.771	0.775	0.781	0.786	0.792
7–16	0.799	0.804	0.807	0.812	0.819	0.820
1–2	0.846	0.850	0.852	0.856	0.858	0.862
5–8	0.919	0.920	0.921	0.923	0.925	0.926
3–4	0.966	0.966	0.966	0.967	0.968	0.969
4–5	0.978	0.978	0.979	0.979	0.980	0.981
9–10	0.995	0.995	0.995	0.995	0.995	0.995

To find the mean pressure exerted throughout the stroke, multiply the absolute initial pressure (above a vacuum) by the number opposite the given cutoff and in the column corresponding to the percentage of clearance, and deduct the absolute back pressure.

For ready use we may write

$$[(p + 15) \times ratio] - 17 = M.E.P.$$

for noncondensing engines, and

$$[(p + 15) \times ratio] - 3 = M.E.P.$$

for condensing engines, in which p is the boiler gage pressure.

Percentage Gain in Power

Example: What will be the percentage of gain in power by connecting a condenser to an engine having a clearance of 5 per cent., cutting off at ¼ stroke, and using steam at 100 pounds gage pressure? The engine is rated at 200 horsepower when running noncondensing, what will be its power when running condensing?

$$[(100 + 15) \times 0.62] - 17 = M.E.P. = 54.$$

Then

$$\frac{1200}{54} = 22.2$$

per cent. gain in power, and

$$200 \times 1.222 = 244,$$

the horsepower that will be developed running condensing.

Fig. 11 shows the effect of a condenser when the power of the engine is kept the same by changing the point of cutoff. The dotted lines in this case show the form and position of the diagram when running noncondensing, and the full lines the diagram when condensing. The areas of the cards are the same in each case, which means the work done per stroke, or in other words, the power of the engine is not changed, but cutoff occurs earlier in the second case, so that the amount of steam used is less. If cutoff occurs at ¼ stroke when running noncondensing and at 1/6 stroke when condensing, the saving per stroke will be the difference (neglecting clearance) between ¼ and 1/6 of a cylinderful of steam. This expressed as a percentage will be

$$\frac{0.25 - 0.166}{0.25} \times 100 = 33$$

per cent.

Saving in Fuel

Let us compute the saving in fuel by using a condenser in connection with a 20x36-inch noncondensing engine running at a speed of 200 revolutions per minute, having 5 per cent. clearance and cutting off at ¼ stroke; boiler pressure 100 pounds gage. The water rate is 32 pounds per horsepower-hour, and the efficiency of the boiler is such that one pound of coal will evaporate nine pounds of water. The method of making the calculations is as follows:

$$M.E.P. = [(100 + 15) \times 0.62] - 17 = 54$$

pounds.

Next find the point of cutoff which will be required to maintain the same power of the engine when running noncondensing. In this case the initial and mean effective pressures remain the same, the only change being in the back pressure, so we may write

$$[(100 + 15) \times R] - 3 = 54,$$

and solve for R as follows:

$$(100 + 15)\, R = 54 + 3;$$

hence,

$$R = \frac{54 + 3}{100 + 15} = 0.49 +$$

Looking in the table of ratios we find that 0.49 corresponds very nearly to a cutoff of 1/6, which may be taken with sufficient accuracy as the new point. The saving in steam per stroke will then be

$$\frac{¼ - ⅙}{¼} = \frac{0.25 - 0.166}{0.25} = 0.33,$$

or 33 per cent.

The horsepower of the engine when running noncondensing is

$$\frac{54 \times 3 \times 3.14 \times 400}{33{,}000} = 616$$

and the weight of steam required is, therefore,

$$616 \times 32 = 19{,}712$$

pounds per hour.

The coal required to generate this weight of steam is

$$\frac{19{,}712}{9} = 2190$$

pounds.

If the saving in steam by running condensing is 33 per cent., then the saving in coal, being in direct proportion, will be

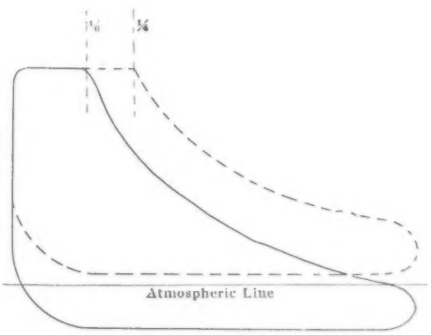


FIG. 11. CARDS OF EQUAL AREA

$$2190 \times 0.33 = 730$$

pounds per hour.

To make the comparison complete, the cost of cooling water must be considered, also the interest charges and depreciation on first cost of the condensing apparatus.

Use of Exhaust Steam a Factor

Another factor entering into problems involving the use of a condenser is the use of the exhaust steam, as for heating during the winter months in a factory or office building. This makes it necessary to raise the back pressure somewhat unless some form of vacuum system is used. In cases of this kind the question is often raised as to whether it will be better to install a condenser and use live steam for heating, or raise the back pressure and run noncondensing, using the exhaust for heating. Sometimes a condenser is installed for summer use only. Here, as in the examples already given, the existing conditions should be carefully studied, and the various items considered, such as cost of coal and cooling water, type and cutoff of engine, relation of exhaust steam to that required for heating, and the length of the heating season, etc. A practical example will show the general method of solving problems of this kind, and can be changed more or less to suit existing conditions.

Let us take the case of a mill where power is supplied by a slow-speed 24x48-inch noncondensing engine running at 100 revolutions per minute, cutting off at ¼ stroke, and having a clearance of 3 per cent. Boiler pressure 90 pounds gage, water rate of engine 28 pounds per horsepower-hour, and evaporation of boilers nine pounds of steam per pound of coal; cost of coal, $3.50 per ton. Cooling water can be had for the cost of pumping only, which amounts to, say, 600 foot-pounds per gallon. Steam is required for heating six months in the year, in sufficient quantity to supply 30,000 square feet of direct radiating surface. Neglecting the cost of the condensing apparatus, will it be more economical to run noncondensing, using the exhaust steam for heating under a back pressure of five pounds, or to install a condenser and pumps and heat with live steam directly from the boilers?

Noncondensing Plant

Let us take first the case of a noncondensing plant. The power developed by the engine under the conditions will be

$$M.E.P. = (90 + 15) \times 0.61 - 20 = 44$$

pounds, and

$$H.P. = \frac{44 \times 4 \times 452 \times 200}{33{,}000} = 482.$$

The weight of steam required per hour will be

$$482 \times 28 = 13{,}496$$

pounds, and the weight of coal

$$\frac{13{,}496}{9} = 1499,$$

or call it 1500 pounds, from which the cost will be

$$0.75 \times 3.50 = \$2.62.$$

In an engine of this type the cylinder losses may be taken at about 20 per cent., which will leave

$$13{,}496 \times 0.80 = 10{,}796$$

pounds of steam available for heating purposes. About one-fifth of this can be utilized for heating the feed water the year around, leaving, say, 8650 pounds of steam for radiator service. One square foot of direct radiating surface will condense about one-third pound of steam per hour under ordinary conditions, from which 30,000 square feet will condense 10,000 pounds. This calls for

$$10{,}000 - 8650 = 1350$$

pounds of live steam per hour in addition to the exhaust, and at an expense of

$$\frac{1350 \times 3.50}{9 \times 2000} = 0.26$$

dollar per hour, making the total cost of fuel

$$2.62 + 0.26 = \$2.88$$

per hour.

Condensing Plant

Next compute the saving in fuel for power by the use of a condenser. The first step in this computation is to determine the required point of cutoff to give the same power with the reduced back pressure.

$$(90 + 15) \times R - 3 = 44,$$

from which

$$105\,R - 3 = 44,$$

or

$$R = \frac{44 + 3}{105} = 0.45.$$

Referring to Table 2 we find this ratio corresponds approximately to a cutoff of 1/7, which will give a saving in steam of approximately

$$\frac{\frac{1}{4} - \frac{1}{7}}{\frac{1}{4}} = \frac{0.25 - 0.14}{0.25} = 0.44,$$

or 44 per cent. This makes the weight of coal required for power alone

$$1500 \times 0.56 = 840$$

pounds per hour, and to this must be added the fuel required for heating and for pumping the condensing water.

The weight of steam required for the heating system was found to be 10,000 pounds, and

$$\frac{10{,}000}{9} = 1111$$

pounds of coal per hour. The feed water can be warmed by the exhaust from the engine and pumps and so does not appear as a separate item. In computing the quantity of cooling water required, it is necessary to determine the terminal pressure at release in the engine cylinder. Looking in Table 1, we find the ratio for 1/7 cutoff and 3 per cent. clearance to be approximately 0.170, which gives a terminal pressure of

$$(90 + 15) \times 0.170 = 18$$

pounds absolute. The total heat above 32 degrees in a pound of exhaust steam is, therefore, 1150 heat units.

The average temperature of the cooling water during the heating season may be taken as 40 degrees, and the discharge water as 115 degrees. Substituting these values in the formula for quantity of cooling or injection water, we have:

$$Q = \frac{1150 - (115 - 32)}{115 - 40} = 14.2$$

pounds per pound of steam condensed.

The saving in fuel and steam by the use of a condenser was found to be 44 per cent., hence the weight of steam is

$$13{,}496 \times 0.56 = 7557$$

pounds per hour, and if 20 per cent. of this is condensed in the cylinder, it will leave

$$7557 \times 0.80 = 6045$$

to be cared for in the condenser. Therefore, the weight of cooling water will be

$$\frac{6045 \times 14.2}{8\,3} = 10{,}342$$

gallons. If 600 foot-pounds of work are necessary for pumping each gallon to the condenser, it will require

$$600 \times 10{,}342 = 6{,}205{,}200$$

foot-pounds per hour, or

$$\frac{6{,}205{,}200}{60} = 103{,}420$$

foot-pounds per minute, which is equal to

$$\frac{103{,}420}{33{,}000} = 3.134$$

horsepower.

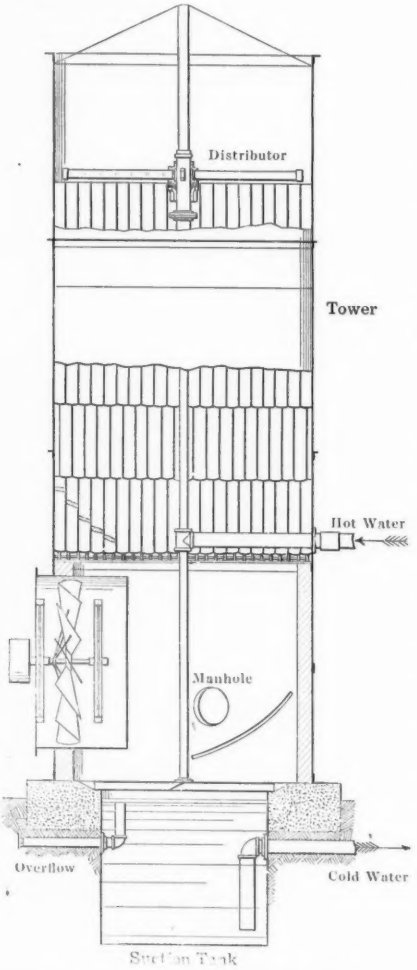


FIG. 12. TYPE OF COOLING TOWER

Small direct-acting steam pumps require on an average 120 pounds of steam per horsepower, from which the weight of steam required for this purpose will be

$$120 \times 3.1 = 372$$

pounds per hour, and requiring

$$\frac{372}{9} = 41$$

pounds of coal. Therefore, the total weight of coal required per hour will be as follows:

Engine	840
Pump	41
Heating	1111
Total pounds	1992

which we may call one ton, and costing $3.50 as against $2.88 when running noncondensing, during the heating season only.

Summary

Assuming the mill to run eight hours per day for 300 days, the weight of coal required when running noncondensing will be

$$8 \times 300 \times 1500 = 3{,}600{,}000$$

pounds for power, and

$$8 \times 150 \times 150 = 180{,}000$$

pounds additional for heating during 150 days, making a total of 3,780,000 pounds per year.

In the second case there will be

$$8 \times 300 \times (840 + 41) = 2{,}114{,}400$$

pounds for power, and

$$8 \times 150 \times 1111 = 1{,}333{,}200$$

pounds for heating, making a total of 3,447,600 pounds for the condensing plant. Then

$$3{,}780{,}000 - 3{,}447{,}600 = 332{,}400$$

pounds, from which we find the difference in cost in favor of the condensing plant to be

$$\frac{332{,}400 \times 3.50}{2000} = \$581.70$$

per year. The above example takes no account of the fact that the temperature of the cooling water will be higher in summer, thus requiring a greater quantity. This will, of course, increase the weight of coal necessary for pumping, and corrections can be made for this if desired.

In any particular case the data furnished will, of course, determine the method of making the computation, but the foregoing will serve to illustrate how some of the more important items enter into the problem, and can be used as a guide by making the proper changes and substitutions. If one wishes to go into the smaller details more accurately, it is a simple matter to do so.

If a vacuum system is used, it affects the computations only so far as the back pressure on the engine is concerned. So much depends upon the tightness of the system of piping and radiators and upon the efficiency of the apparatus used that it is impossible to state the amount of vacuum which can be maintained in this way. In practice it is customary to obtain from the makers of the system it is proposed to use, a guarantee covering the back pressure which they will agree to maintain under the stated conditions, and use that as a basis for computation.

Reducing Cost of Cooling Water

A very important factor affecting the value of a condenser is the cost or availability of water for cooling purposes, as already stated. In order to reduce the cost in localities where water is scarce, or the expense great, various methods have

been adopted for cooling it after passing through the condenser and using it over again. The cooling tower, so called, is the device most commonly used for this purpose. There are several forms of this in use, the general principles being practically the same in each case.

The discharge water as it leaves the condenser is pumped to the top of the tower, where it is distributed over a series of extended cooling surfaces. As it falls over these in thinly divided streams or sheets, it meets an upward current of air which is produced by a fan or by the flue or chimney effect of the inclosing shell of the tower. The cooling of the water is accomplished in three ways: First, by radiation from the pipes and sides of the tower; second, by the contact of the water with the cooler air; and third, and by far the most important, the effect of evaporation.

It is claimed that with the best forms of cooling tower the water can be reduced in temperature 40 or 50 degrees, the loss from evaporation not being greater than 3 to 4 per cent. of the total quantity of water passing through the tower. The Worthington tower, shown in Fig. 12, illustrates the general form, where a fan is used for forced circulation of air.

In this case the cooling surface is made up of pieces of 6-inch tile pipe about 2 feet in length, and placed on end, as shown, where the casing is cut away. The whole is inclosed in a steel-plate case, and a circulation of air is produced by means of a disk fan at the bottom. The discharge water is pumped to the top of the tower, where it is discharged through a special distributing device over the upper course of the evaporating surface. From here it falls slowly over the successive

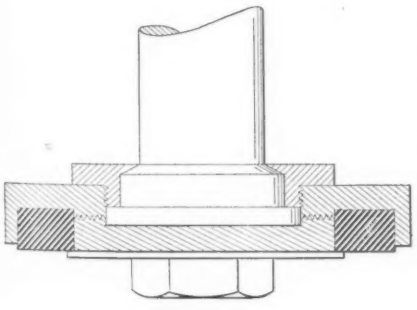


FIG. 2

courses, to the cold-water tank at the bottom.

Natural-draft cooling towers are similar in construction except the fan is omitted, and the top of the shell is extended in the form of a flue for a considerable distance above the cooling surface. The tower may be placed just outside the boiler house, or upon the roof of the building, as most convenient.

Locating and Adjusting Valves

By W. H. Wakeman

The vertical boiler shown in Fig. 1 is fitted with a Y-valve on the blowoff pipe and is located so as to shut against the pressure; therefore, the packing nut can be removed at pleasure whenever the

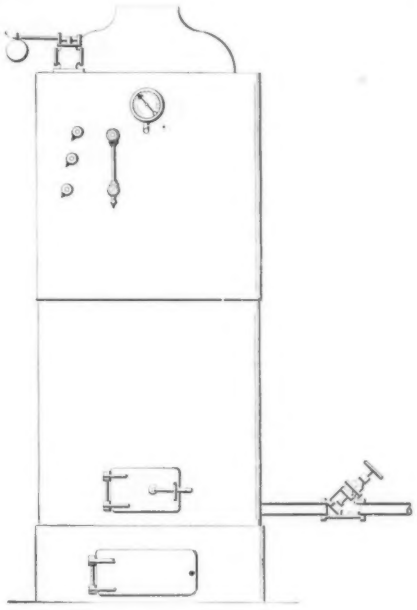


FIG. 1

valve is shut. One day this valve failed under about 110 pounds pressure, giving the fireman a lively job to haul the fire before the water level fell below the crown sheet, as it is not a small boiler.

As soon as possible the bonnet of the valve was removed, showing that the disk holder was broken into two pieces, as illustrated in Fig. 2. When the valve stem was screwed down hard the outer part of the disk holder was supported by the valve seat (which is not shown in the cut), but the central part was forced downward, making a clean break as indicated by the rough lines just inside of the disk on both sides of the stem. The metal appeared to be in good order, and the casual observer would not hesitate to say that there was enough of it to be safe, when other parts of valves are considered in comparison; but an illegitimate strain may have caused the failure.

This allowed the ring shown in Fig. 3 to be forced up into the bonnet, leaving the valve wide open. This ring is in true circular form, both inside and outside, because these surfaces are machined, while the break is represented by the shaded part of the cut.

The illegitimate strain mentioned is sometimes caused on these valves by careless firemen, who remove the bonnet when cleaning the boiler, in order to ascertain if sediment has collected at this point. When replacing the bonnet the stem is not always run back as it ought to be, but is turned forward enough to bring the disk in contact with its seat before the bonnet is down firmly in contact with the body; consequently, when the bolts are in place, or the bonnet screwed down, as the case may be, a very great stress is brought to bear on the disk holder.

Undoubtedly this is what caused the failure in this case, although the actual rupture did not occur until change of temperature resulted in unequal expansion of the parts, and the effects of pressure increased the strain which became too great for the metal composing the disk holder.

This accident provides an argument in favor of reversing the position of such a valve, thus bringing pressure on top of the disk, which would hold it to its seat in case of accident, instead of blowing it away. However, when a valve is so placed it is much less convenient to operate, because all lost motion in the threads and disk holder must be disposed of by a pull on the stem when an effort is made to open it; then, as soon as the disk rises and pressure accumulates under it, lost motion is suddenly taken up in the opposite direction; consequently, whatever opening is thus secured comes with a snap that is not safe, because it greatly increases the strain on all joints affected by the action thus produced. When located as shown in Fig. 1 all lost motion is taken up in one direction and there is no reverse action, and it can be controlled much better. Another point in favor of this plan is that the valve stem can be packed at any time without removing pressure from the boiler.

The Use of Gate Valves

As a general rule gate valves are pre-

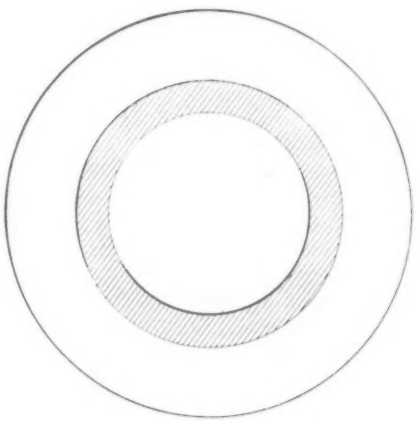


FIG. 3

ferred for this service, but they do not give satisfaction in every case. Fig. 4 illustrates one of these valves located with the stem in a vertical position. Assuming that this valve is on the blowoff pipe of a steam boiler, when the gate is raised to allow scale and sediment to be washed along the bottom of the pipe, some of it will lodge in the pocket *A* and prevent

the gate from descending into the position shown; consequently, the valve leaks and efforts to stop it by forcing the stem downward are not successful.

When the bonnet is removed, the gate lifted and the sediment taken out, the valve may be tight again unless the action of water leaking through has injured the seat, but it is not always practical to do this, because all pressure must be removed and the water allowed to run out before the bonnet can be taken off. If the piping is arranged so that this gate valve will be located in a vertical piece, sediment cannot lodge in the pocket *A* and the gate will never be prevented from seating properly on this account.

Fig. 4 shows the packing nut *B* unscrewed and lifted up to the wheel. The gland *C* is raised out of the stuffing box *D*, but this shows how these parts ought to be proportioned, rather than the condition which usually confronts an engineer when he wishes to repack the stem of this kind of a valve. In this case it is convenient to lift the gland *C* into the nut *B*, thus affording plenty of room to insert a hook into the stuffing box *D* and remove all of the old packing, as none of it should be left there. However, in many of these valves the stem is so short that the gland cannot be raised higher than it is now, as it would already be in the nut and the latter would be against the wheel which cannot be easily removed, because it is

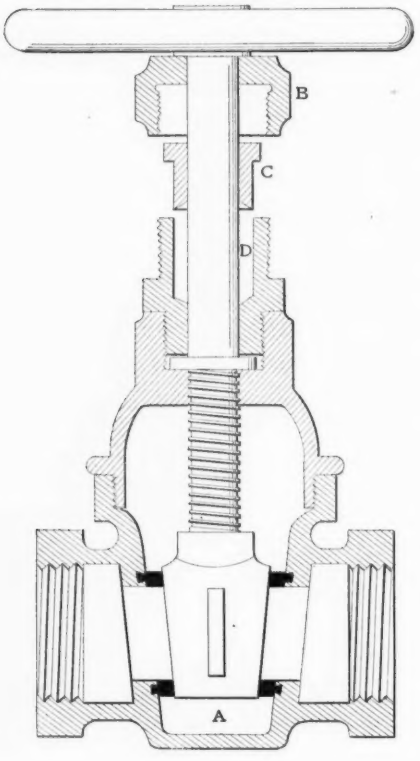


FIG. 4

fastened with a pin instead of being held on by a small nut on the end of the stem.

How does the manufacturer of such a valve expect properly to pack the stuffing box? The designer has evidently never attempted to do this, especially when it is located where it is almost impossible to manipulate a wrench or handle a packing hook. Many times while engaged in this work, which took about four times as long as it would have done with a better designed valve, I have most earnestly wished that whoever was responsible for such an inferior design might be compelled to pack them ten hours per day for at least one month, as that experience would be a valuable teacher, and posterity would be benefited thereby.

In many cases engineers have found these valves with little or no packing in the stuffing box, but the gland was down as far as it would go, then packing had been put into the nut and it was screwed down the same as if there was no gland on the stem. Such an apparent blunder is probably due to ignorance on the subject, but it may possibly have been caused by inability to get the packing under the gland where it belongs.

Another Type of Valve

Fig. 5 is an external view of a valve made similar to the preceding, but not exactly like it. The hexagon part of the bonnet is so small that it is practically impossible to remove it with a monkey wrench, for as soon as force is applied to the handle the jaws round off the brass corners without starting the joint. If this part was made larger the bonnet could be removed without disfiguring the finished work. I have seen valves on which a Stilson wrench has been used for this purpose and it certainly appeared to be a most barbarous practice at that time, but later experiences have modified that decision.

After a monkey wrench has been applied to a hexagon bonnet, in three different positions and spoiled the corners on all of them without breaking the joint, the valve looks as dilapidated as could be expected, and any other kind of a wrench could not make it look worse. I use an old 24-inch Stilson wrench with dull teeth for removing these bonnets. It never fails to do the required duty and when the bonnet is unscrewed, there are two places which show the prints of these teeth. The use of a file with one safe edge soon removes all evidence of rough treatment and the job is finished, leaving four corners that have not been touched, and two that are finished equal to or better than new.

The outward appearance of this valve shows that the gate is made in the form of a wedge. The right-hand or square side side of the body ought to be located in the direction from which pressure acts as the best joint is made here between the gate and the seat. If the valve is tight at this point as it ought to be, the stuffing box can be repacked whenever the valve is closed, provided it is put on the pipe as above described and pressure acts on one side of it only. These remarks concerning gate valves refer more especially to those that are designed for 2-inch pipe

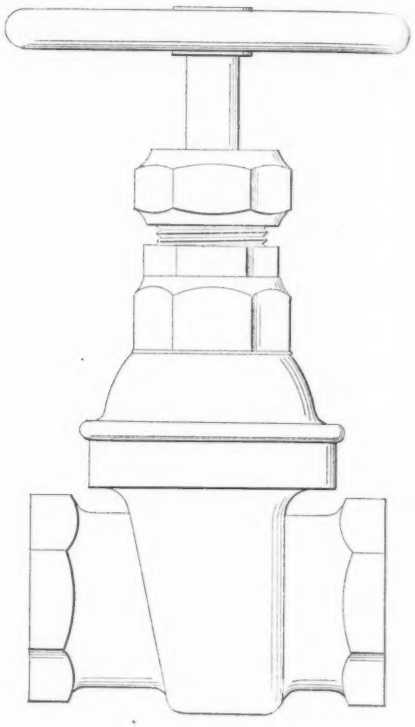


FIG. 5

and smaller sizes, as the larger sizes are usually given more generous proportions.

Annual Report of the Chief of the Bureau of Steam Engineering

Admiral John K. Barton, engineer-in-chief of the United States Navy, has just issued his report as Chief of the Bureau of Steam Engineering. The bureau has taken advantage of every opportunity to obtain information regarding the use of internal-combustion engines, but has evidently not seriously considered anything other than the smaller sizes. A number of motors of different designs have been purchased and are now in service, but have not proved entirely satisfactory, due, it is believed, to unfamiliarity on the part of the enlisted force in operating motors of this class. An internal-combustion engine, built at the Norfolk navy yard, has been fitted in a sailing launch. It is four-stroke-cycle with two 4¼x5-inch cylinders, develops 10 brake horsepower and has demonstrated the value of such an installation in launches having great carrying capacity at slow speed.

The three scout cruisers, "Birmingham," with reciprocating engines; "Chester," with Parsons turbines, and the "Salem," with Curtis turbines (the boilers of the "Chester" being Normand and those of the "Birmingham" and "Salem" modified Normands), have been completed and most satisfactory trials made by all of them.

Measuring tanks, pumps, pipes, etc., are being fitted on each of these vessels and as soon as completed, exhaustive trials will be made to determine accurately the steam consumption of each type of propelling machinery and the endurance of the three vessels at sea under the same conditions.

The bureau is standardizing as rapidly as possible all details where this can be done to advantage and considers that it will result in a less number of different designs, the only reason for which being that the contractor had a pattern. By the use of standards, as of grate bars, for instance, the spares could be used on several vessels instead of each vessel requiring a different one.

At the Engineering Experiment Station at Annapolis, the work of installing the experimental 500-brake horsepower Parsons marine steam turbine was completed in December, 1907, and a series of tests was made at intervals during the winter. A notable fact is that the marine turbine was erected, adjusted and run after a few trials immediately up to full power by men none of whom had any previous experience with it and only one of whom had ever seen such a machine in operation. This is due to the inherent simplicity of the machine, the construction of which admits of all the finer adjustments being done in the builder's shop; just the reverse of the reciprocating engine, where the fine adjustments must be made by the force on board after installation and a number of trial runs.

Another fact observed was that although the new boiler foamed badly, the turbine was not injured in the slightest degree. The only evidence of the foaming was that of the calorimeter, the leaking of water from the gland between the high-pressure and low-pressure turbines and the excessive steam consumption for the power developed. The foaming was probably due to carrying 150 pounds pressure on a boiler designed for 500 pounds and was corrected by wiredrawing from 200 pounds down to 150, when it was found that the weight of dry steam used per brake horsepower became normal and as guaranteed by the builders, that is, 16 pounds.

Plans have been laid out for the following tests during the coming year, most of the apparatus for which is on hand or about to be delivered: Parsons marine turbines, two types of surface condenser, fire-room blowing engines, CO_2 recording apparatus, coal calorimeters, corrosion of metals, lubricants and bearing metals, packing for steam and water joints and boiler-water gages. For the next fiscal year, it will undoubtedly be found desirable to continue the tests of turbine engines and condensers, the subject being too large to be exhausted in one year. In addition, types of distilling apparatus, feed pipes, air pumps, circulating pumps, feed heaters, oil burners, mechanical stokers, safety valves, and other boiler and engine attachments and fittings should be tested with a view to selecting the most efficient as a standard for naval use.

The report strongly urges the instruction of young line officers in engineering, and maintains that engineering in the navy has now reached such a point that two things must be done. For temporary relief, at least twenty officers of the grades of lieutenant commander and lieutenant should be ordered at once to shore duty connected with the steam-engineering bureau, and a fixed policy should be adopted of assigning permanently a certain number of officers to engineering duty only, such assignment to begin upon entering the grade of lieutenant and officers so assigned to perform shore duty only after reaching the grade of commander.

Ontario Engineers Must Have Licenses

After January 1, 1909, each stationary engineer in the province of Ontario, Canada, must have a license if he wishes to operate a plant of over 50 horsepower. The license act was passed April 20, 1907, and it has been recently announced that the Board of Examiners has been fully appointed, consisting of W. C. McGee, of Toronto, chairman; W. B. Varley, secretary; and two practical engineers, namely, Charles Moseley, chief engineer of the Toronto Electric Light Company, and William Corrigan, chief engineer in the School of Physics, Toronto. It is expected that nearly 4000 engineers will be examined before the first of the year. Engineers who have been in charge of a steam plant of 25 horsepower or over in Ontario, who held a certificate granted either by an association of stationary engineers in Ontario, or a Canadian marine certificate, or were qualified locomotive engineers at the time the law was passed, or can satisfactorily show that they have had at least two years' experience in Ontario in charge of a steam plant of 25 horsepower or over previous to the passing of the act, April 20, 1907, are exempt from an examination and may secure a license by filling out certain forms prepared by the board. All other engineers must be examined, and these examinations may take place, subject to appointment, at any time during office hours at the office of the board at the parliament building, Toronto, and in addition the board has the privilege to conduct examinations twice a year at the cities of London, Kingston, Ottawa and Port Arthur, or at other places if it is considered necessary. Candidates must present testimonials as to character and sobriety before they are permitted to take the examination, and in the examination must answer correctly at least 60 per cent. of the questions presented by the board and approved by the Minister of Agriculture, except when given an oral examination, which is left to the discretion of the board. The fee for examination, including certificate, is to be $3; for an initial certificate granted by an oral examination, $2; for renewing a certificate or issuing a duplicate certificate, $1, and for the renewal of a certificate where the holder has failed to register before the first day of February, $5.

A "Skyscraper" Plant

On Thursday afternoon, December 3, the students of Stevens Institute were afforded the pleasure of hearing a lecture on the construction of "skyscrapers," given under the auspices of the Engineering Society. The speaker of the afternoon was Charles G. Armstrong, the architect of the Singer building, New York, and his treatment of the subject was limited more particularly to a consideration of that structure. Starting with a definition for "skyscraper" and the necessity for its erection, Mr. Armstrong immediately digressed to the subject of elevators, clearly pointing out the fact that the modern skyscraper was almost totally dependent for its usefulness upon the modern elevator.

Special attention was paid to the manner of anchoring the massive tower of the Singer building. In this connection Mr. Armstrong stated that the feet of the tower columns were secured to the foundations in the subcellar by stout steel tie rods running down 40 feet in the concrete, and that the enormous lifting force of 250,000 pounds per square inch would be necessary to budge a column.

After a short description of the various safety devices and signal apparatus, shop facilities for repair work, etc., Mr. Armstrong took up the consideration of power. He stated that the Singer building had as much power as a moderately sized city, 2000 horsepower roundly. Boilers, engines and electrical apparatus of highest type securing economy, convenience, and safety adequate to wide possibilities of accident and extremes of load testify to the disregard of expense in attaining perfection, the final testimony to the Singer company spirit being the great external lighting system of the tower, used for advertising purposes.

In closing his highly instructive lecture Mr. Armstrong made an appeal to technical men to take up as a vocation the operation of great power plants and buildings with a view of improving upon the wasteful methods of running them today.

The American Society of Mechanical Engineers has established its first student section by affiliating the Stevens Engineering Society with the national society. Though the undergraduate associations retain their personnel and autonomy, it is expected that both the allied societies as well as the A. S. M. E. will benefit from this closer contact of the practical engineer and the engineer-student.

POWER AND THE ENGINEER

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The Engineer and the People

Morris L. Cooke injected a new line of thought into the recent meeting of the American Society of Mechanical Engineers by urging upon the members a sense of their responsibility to the people. The thought advanced by President Hadley, of Yale University, in his address at the dedication of the Engineering Societies' building in May of 1907 that: "There are three professions today which do not regard themselves as servants, but as masters, the financier, the journalist and the politician. If the engineer and the lawyer accept positions as servants, simply putting their technical knowledge at the disposal of the merchant, journalist or politician who will pay the highest price for it, it is not simply a question of inferiority, but of dereliction of public duty," was taken up and expanded.

No one who gives the matter thought will be likely to contest the assertion that in the manifold activities of the society and of the profession the general public as such has been almost ignored. Among the announced committees of the society not one has a title indicating a function broader than the conservation of the interests of engineering and of the engineering profession. Among the thousand and more papers listed in the last published index to the *Transactions* it is difficult to find a single title which indicates that the author was addressing anyone beyond his professional audience. Inquiry among the lay public will show that while the society is given full credit for its professional strength, absolutely no credit is given to it for interest in public matters. Mr. Cooke suggests, and the suggestion was adopted by the society, the appointment of a Committee on Relations with the Public, which will seek to give publicity to the fact that the American Society of Mechanical Engineers stands ready to offer disinterested advice through its council to Government officials, municipal, State and national, similar to the relations now existing between the Federal Congress and the National Academy of Sciences; to establish such relations with the lay press as would make its advice and help sought when engineering matters are up for public discussion; to provide for courses of lectures for the general public, to be given under the supervision of the committee in the Engineering Societies' building. Two ideal lectures of this kind were given during the recent meeting; one by Professor Brashear upon "Photographing the Stars," and the other by Lieut. Frank P. Lahm, U. S. A., upon "Aëronautics."

As an instance in which the mechanical engineer could have been of immense service to the public, an investigation of the fire risk in theaters, conducted by John R. Freeman, a past president of the society, following upon the Iroquois theater fire at Chicago, is cited. The investigation was made at the expense of a public-spirited man "not for the purpose of fixing the blame, but to help us find out how such fearful disasters can be prevented." Mr. Freeman's report contains specific recommendations as to theater construction and the prevention of fires in buildings of this class, which were so logical and relatively so simple that if they could have been made of general public knowledge theater owners and proprietors must almost inevitably have been forced by public opinion to their observance, if not even impelled thereto of their own accord. And yet all the publicity that was given to this work was through the distribution of copies printed at private expense to theater proprietors and the officials of a few municipalities.

Just as the medical profession has learned to treat disease broadly, to work together for its prevention, suppression and eradication, rather than to potter individually with specific cases and to guard successful prescriptions and treatment as professional secrets and private property, so the engineer must arise to a broad conception and collective treatment of the great problems of engineering with which the age is replete.

The records of engineering in the early days of the last century show that engineers had not then learned how to cooperate with each other. Up to a comparatively recent date, the professional knowledge of an engineer was his own property, and he felt that in imparting any of this knowledge to another engineer he was doing himself a decided injury. Discussions between engineers were carried on mainly with the idea of doing each other harm rather than good. The large number of engineering societies, with the fundamental object of the free exchange of engineering data between their members, shows how thoroughly engineers have learned that the general policy of giving freely of their knowledge to their fellow practitioners increases rather than diminishes their effectiveness and, in proportion, their earning power.

In proposing the appointment of this new standing committee Mr. Cooke suggests only the extension of this coöperation to include the public. There is good reason to believe that engineers have as much to gain from coöperation with the public as they have undoubtedly gained from coöperation with each other. The time may come when the increasing number of papers of interest alike to the engineer and the layman will suggest the necessity for a Section on Public Matters. It is no longer possible for either a profession or a craft to corner information and hold it for its own use. Broadly speaking, those who seek information in any field can obtain it, or at least enough to answer their immediate purposes. This danger would in itself constitute a sufficient reason for the engineer to take the

public into his confidence. For, after all, it is public opinion and not the dictum of the engineering fraternity which finally decides the large questions of engineering practice. Only by educating the public to understand and appreciate the work of the engineer can the public be made to demand the best that can be devised and executed by trained and skilful men. This will have a twofold beneficial effect on the profession, in that it will make more work for the engineer and will give the public that general acquaintance with engineering matters which will make it suspicious of short cuts.

Carelessness Leads to Accident

Not long ago a new engine was installed in a certain plant to replace a smaller unit. The new machine called for a belt six inches wider than the old one, and this difference of half a foot was the cause of an accident fatal to the night engineer. It had been the custom to step over the old belt while it was in motion, but the additional width of the new belt made this procedure so hazardous that orders were given by the chief engineer to discontinue the practice. The night engineer received a written order to this effect and was told of the danger, but with careless disregard for his own personal safety, he took the fatal step and later was picked up in pieces.

This is one of the many instances that might be enumerated to show how carelessness leads to accidents. The International Congress for Prevention of Accidents has listed many more, and new cases are continually arising. In the particular case cited, the personal safety of the operator was the only question immediately involved, but a man careless of his own personal safety is inclined to be even more careless of the safety of others and of the machinery under his charge. These are the men who fall asleep while on duty, fully intending to awake before the water drops out of sight in the gage glass or a bearing burns itself out for lack of oil.

There is no place for such men in the power plant. The operator of boilers and engines should be alive to the dangers in the plant, and above all things should be careful and exacting in his work. It is not essential that he should know all about the design and construction of the apparatus under his charge, but he should know all about its operation, where the danger points are and how to avert them. When perfectly familiar with all that is essential to know about the safe operation of the plant, eternal vigilance should be the watchword, for it is only by exact and painstaking attention to detail that accidents can be eliminated, or, at least, their number curtailed.

In examining engineers as to their fitness for operating plants, their general character should have an important bearing, and it should be ascertained beyond doubt that they have inherent qualities which tend to make them careful and trustworthy.

Reinforcing Manholes

The article by S. F. Jeter on another page is the first complete treatment of the reinforcement of openings made in boiler shells for manholes, etc., which we have seen. The subject is ordinarily dismissed with directions to "make the section of the reinforcing ring equal to the section of metal removed." Mr. Jeter's directions include those for riveting the ring as well as for determining its width and thickness, whether it be made of the same material as the shell or of cast iron. The treatment is exceedingly simple and no one, who does not try to read the article backward and gets stuck upon the formulas before he understands what they mean, will have any difficulty in following and applying it.

Publicity, Why Not?

Considerable has been published in these columns pertaining to blunders in power plants. Such articles have criticized chiefly so-called "mechanical engineers," and the comments have not always been to their credit. Recently a letter was received from one who had been thus criticized, who complained that certain published statements concerning his work should have been left unpublished. He said: "It is a reflection on the great body of the engineering fraternity to let such matters get into our journals."

While we sympathize with the gentleman in question, we cannot agree that publicity of blunders made by engineers, mechanical or otherwise, reflects on the engineering fraternity. It is a reflection on the ability of the individual, to be sure, but not the body as a whole. But, suppose it were? Mechanical engineers are no more exempt from making blunders than others. Some are proficient, some are not; some have been trained for the calling, others have stumbled into it; some were born mechanical engineers, while others have picked up their meager stock of knowledge from various questionable sources.

Publicity is a far greater force for inducing people to observe the law than fear of the consequences of breaking the law. So, in the matter of mechanical-engineering blunders, publicity has a wholesome effect, in the direction of avoiding future blunders. A man seldom makes the same mistake more than twice. If he does, it may be justly termed a habit. By throwing the light of publicity on these mistakes they are brought to the attention of others who might otherwise commit similar blunders. In consequence, a large number are benefited at the expense of one.

Surely anything that will cause men to be more thorough and accurate in the execution of their work cannot but find favor with those who would produce nothing but the best they are capable of. Publicity by all means—but let it be tempered with justice.

What Would You Do?

It has been asked editorially and in the correspondence columns what an engineer would do if requested to operate a boiler which appeared to be unsafe. Would he, because the boiler had not up to that time failed, take the chance that it would not while under his care? Would he, because he and those dependent on him needed bread, reason that because that which looked very probable had not happened it would not happen, and put an end to his need for bread as well as for all the other necessities which the term "bread" implies?

Hundreds of boilers have exploded, resulting in some cases in loss of life, but the instances are rare in which it afterward transpired that an unsafe condition was even suspected, much less known to exist.

One correspondent pictures the unhappy predicament in which the man will be found who unsuccessfully tries to persuade the owner that his boiler is unsafe and appeals to the inspector to prove his contention. It is gathered from this that the average owner desires to remain in ignorance of the true condition of his steam plant, particularly if this condition is at all questionable, and that an owner would prefer to have dangerous apparatus in operation rather than go to the slight inconvenience and expense involved in an investigation.

It is stated that the man who is discharged for calling attention to possible dangers and insisting that probable ones be remedied will be voted a "pestilent fellow," classed with other agitators and blacklisted by all owners desiring peace and ignorance. The picture is hardly true to life. One well known engineer has for years refused to work in or even visit a plant where lap-seam boilers were used. So far from being regarded as a general nuisance, his services are at a premium and his advice is sought by owners and operators alike.

But what would you do if brought face to face with the proposition of choosing between the operation of apparently unsafe boilers and hunting for another job?

Engineers are not merely those who know how to shovel coal, set the valves of steam engines, keep a proper water level and the ten thousand other details of steam power-plant operation. They are men, and when it comes to the operation of supposedly dangerous boilers they will not do it. If you are an engineer and this proposition comes to you, you will turn it down.

Book Reviews

VALVE SETTING; SIMPLE METHODS OF SETTING THE PLAIN SLIDE VALVE, MYER CUTOFF, CORLISS AND POPPET TYPES. By Hubert E. Collins. Hill Publishing Company, New York. Cloth; 200 pages, 6x9 inches, illustrated. Price, $2.

This book was written and compiled for the use of supervising, erecting and operating engineers, who have long felt the need of a work containing simple, practical directions for setting the valves of all of the standard engines. As the fundamental principles of valve-gear design are involved in the slide valve, the first part of the work is devoted to the study of this valve and its movement. Then follows a discussion of the Myer cutoff and the releasing gear of the Corliss type. General rules for finding crank and eccentric positions on all classes of reciprocating engines are given in clear, concise terms, which cannot be misunderstood, and which will be found to be of great value to anyone seeking for the reasons in each detail in the operation of valve setting. These rules will enable a man to apprehend the underlying principles and intelligently set any valve, whether described in detail or not.

Much of the matter is compiled from contributions to POWER and has been submitted to and passed upon by the builders of the various engines mentioned. There are twenty chapters, eighteen of which deal entirely with steam-engine valves and their setting. Chapter 19 treats in an exhaustive manner of both simple and compound duplex-pump valve setting. The twentieth and last chapter contains directions for setting the mechanically operated valves of blowing engines and air compressors. Illustrations are numerous and consist principally of line drawings, although a few half-tone reproductions of photographs are used. The work is thoroughly indexed.

MACHINE SHOP CALCULATIONS. By Fred H. Colvin. Hill Publishing Company, New York. Cloth; 172 pages, 4½x6¾ inches. Price, $1.

This book is a clear and simple explanation of all the calculations necessary to the solution of any practical machine-shop problem. It is suggestive of the shop, and the author, who is an expert machinist, tells in plain, everyday English how to do some of the things he has done. The terms used are shop terms and apply to shop work exclusively, from figuring the area of a sheet of iron to the calculation of compound gears for cutting fractional threads.

The twenty chapters of the work are separated into two parts: The first is given over to general shop calculations, such as common and decimal fractions, cancellation, proportion, percentage, square and cube root, mensuration of angles, surfaces and volumes, etc.; the second part is devoted to special calculations which include cutting speeds for lathes, planers, shapers and grindstones, tap drilling and taper work, gearing and pulley sizes, speeds, etc. The work is the applied mathematics of the machine shop adapted to the man and the conditions that the shop develops. The indexing is complete.

PUMPS, TROUBLES AND REMEDIES. By Hubert E. Collins. Hill Publishing Company, New York. Cloth; 99 pages, 4½x6¾ inches. Price, $1.

This is one of the POWER handbooks prepared by Mr. Collins for the use of the operating engineer. The first five chapters deal with pump troubles and the cases cited cover nearly if not quite all the possible sources of pump trouble and the remedies. One chapter is given over to the description and illustration of some novel cases of pump repair and shows how fertility in resource in emergency is a valuable qualification in a man in charge of steam apparatus.

Two chapters tell of the methods of valve setting on duplex pumps, both simple and compound. There are also chapters upon boiler-feed pumps and horsepower of pumps in general, on pumping tar, molasses, cocoa liquor and other heavy liquid and semi-liquid substances. There are tables of pumping-engine performance and tables of various kinds of useful information. One chapter gives general directions for erecting and operating steam pumps. The work was prepared by an engineer from his experience which covers nearly twenty years of active service in engine and boiler rooms. The illustrations and index are comprehensive.

SHAFT GOVERNORS; CENTRIFUGAL AND INERTIA; SIMPLE METHODS FOR THE ADJUSTMENT OF ALL CLASSES OF SHAFT GOVERNOR. By Hubert E. Collins. Hill Publishing Company, New York. Cloth; 120 pages, 4½x6¾ inches, illustrated. Price, $1.

This is one of a series of handbooks brought out by the Hill Publishing Company that should find a place in the library of every operating engineer. It consists principally of a compilation of matter that has appeared in the reading columns of POWER, with the addition of some special articles which have been prepared to make the work a complete handbook on the subject.

Twenty-four pages in the first part of the work are devoted to a history and description of the evolution of the shaft governor. Then follow definitions and rules relating to the adjustment of the different types of governor in use in this country. Chapter 12 relates to the Curtis steam-turbine governor, and the last chapter is devoted to instructions for changing the speed of engines controlled by pendulum governors. There are thirty-five line-drawing illustrations and a complete index.

STEAM BOILERS. By Peabody and Miller. John Wiley & Sons, New York and London. Cloth; 419 pages, 6x9 inches; 176 illustrations. Price, $4.

Upon its appearance in 1897 this book took a high position in the literature of its subject. Cecil H. Peabody is professor of naval architecture and marine engineering and Edward F. Miller of steam engineering, both at the Massachusetts Institute of Technology. The book has been subjected to a general extension rather than a revision and the additions are distributed throughout the work. A short chapter on Superheaters has been added, also a number of tables giving the dimensions of and floor space occupied by the different types of boiler and economizer. The treatment of steam piping has been especially extended.

Books Received

"Electric Motors." By Norman G. Meade. McGraw Publishing Company, New York. Cloth; 159 pages, 4½x7¼ inches; 126 illustrations; tables; indexed. Price, $1.

"Electrical Illuminating Engineering." By William E. Barrows. McGraw Publishing Company, New York. Cloth; 210 pages, 5x8½ inches; 135 illustrations; tables; indexed. Price, $2.

"The Design of Highway Bridges." By Milo S. Ketchum. The Engineering News Publishing Company, New York. Cloth; 544 pages, 6x8¾ inches; 309 illustrations; numerous tables; indexed. Price, $4.

"Mechanical Drawing and Elementary Machine Design." By John S. Reid and David Reid. John Wiley & Sons, New York. Cloth; 439 pages, 5½x9 inches; 329 illustrations; plates; indexed. Price, $3.

"Temperature Entropy Diagram." By Charles W. Berry. John Wiley & Sons, New York. Second edition, revised and enlarged. Cloth; 299 pages, 4½x7¼ inches; 109 illustrations; tables; indexed. Price, $2.

"The Mechanical World Pocket Diary and Year Book for 1908." Published by Emmott & Co., Ltd., Manchester, England. Cloth; 298 pages, with diary and memoranda for 1909, 3¾x6 inches; illustrated; tables; indexed. Price, 50 cents.

The Birmingham (Ala.) Association of Mechanical Engineers held its annual meeting Saturday evening, December 5. After the business session there were a reception and a musical entertainment. On December 17 the members listened to a paper by A. H. Hiorns, principal of the metallurgical department of the Birmingham Technical School, on "Anti-Friction Metals."

Air-pump displacement should be 1/75 of the low-pressure piston displacement on a compound engine.

Foos Vertical Gas Engine

The Foos Gas Engine Company, of Springfield, Ohio, has acquired the designs, patterns and other shop equipment of the Marinette Gas Engine Company and taken up the manufacture of the Walrath vertical gas engine formerly built by that company. The engine is of the multiple-cylinder single-acting type, two cylinders being used on the smaller sizes and three on the larger. Fig. 1 illustrates the three-cylinder construction and the compact arrangement of a direct-connected generating unit for electric light or power service.

The base, which constitutes the crank case, is a single-piece casting with handholes in both sides, giving convenient access to the crank shaft, with light admission through the side opposite the worker. The main shaft bearings are individual castings and can be removed without taking the crank shaft out. The

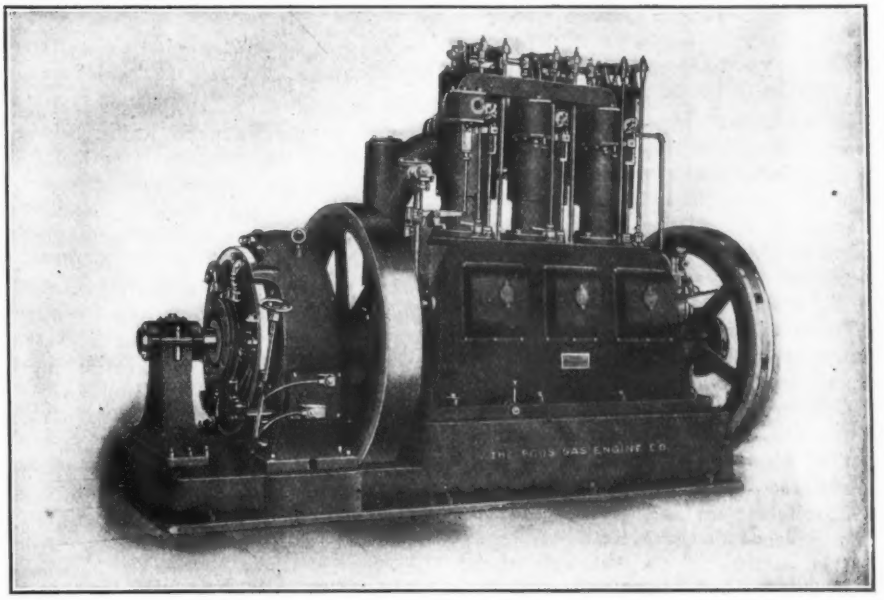


FIG. 1. FOOS THREE-CYLINDER VERTICAL DIRECT-CONNECTED ENGINE

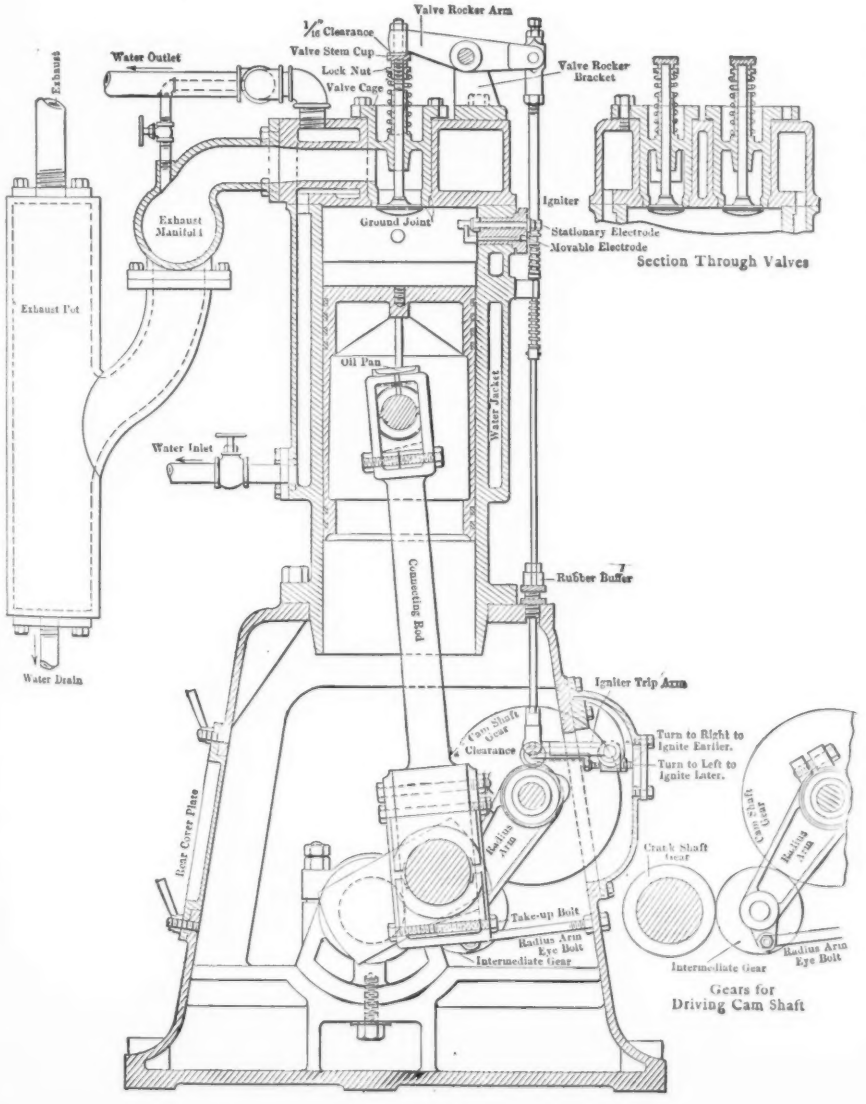


FIG. 2. SECTIONAL ELEVATION OF ONE ENGINE

seating of a bearing in the base frame is shown in Fig. 2. The crank shaft is a one-piece drop forging, made from open-hearth steel, carrying the counterbalances on the rear of the crank cheeks. The connecting rods are six crank lengths long, with adjustable boxes at both ends.

The cylinders are bolted to the top of the crank case and their lower ends fit snugly into the holes in the frame. The valves are mounted in cages set in the cylinder heads, as illustrated in Fig. 2, and are operated through the usual system of rocker arms and push rods by cams on the half-speed shaft. The latter is located in the crank case and driven through an intermediate spur gear, as shown in Fig. 2. The cam-shaft bearings are mounted in the ends of radius arms which are pivoted concentrically with the intermediate gear. An adjusting rod is led out through the side of the crank case and nuts are provided to adjust and hold this rod; by this means the position of the cam shaft relative to the push rods can be adjusted finely to take out all lost motion. The igniters are of the simple make-and-break type operated by push rods, and provision is made for adjusting the time of ignition.

The governor is of the fly-ball type, located outside the crank case, as shown in Fig. 1. It controls a balanced mixing valve opening into an intake manifold or header, and regulates the speed by varying the quantity of mixture admitted to the header. The quality of the mixture is adjusted by hand.

Lubrication of the entire engine is effected by the splash method, the oil in the crank case being the only supply provided

The minor details of construction are clearly shown by Fig. 2 and require no further explanation.

Leather Belting Compared to Rope Drives

By John H. Damon

In reviewing Harrington Emerson's paper on "Leather Belting Compared to Rope Drives," which was read at the annual meeting of the Leather Belting Manufacturers' Association, and was published in Power and The Engineer for December 1, an acceptance of the views as given would result in two conclusions:

That the advocates of rope driving assert their method of transmission to be superior to all others.

That rope transmission is, in every case where it is applied, inferior to some other method which could be adopted; in other words, that rope transmission need not be considered in the solution of any power-transmission problem.

It is needless to say that both of the above conclusions would be incorrect. Regarding the first one, the conservative rope manufacturers have always carefully refrained from advocating the use of rope, except in places to which it is adapted, although, in considering such places, the peculiar merits of rope are generally described in a form similar to that indicated in the quotations given by Mr. Emerson. These merits, however, are not set forth as against those of *all* other methods of transmission, as seems to be assumed by the author of the article under discussion, but are used more particularly in comparing rope with its close rival, leather belting. It must be generally conceded that no one method of transmission has *every* advantage in *every* case; and rope cannot fairly be discriminated against by selecting one advantage from this agent, one more from another, and so on, and in the end summing up the total in a balance against rope.

The claims of the rope manufacturers as stated by Mr. Emerson are as follows:

(1) That the pulleys are narrower for the same power transmission.

(2) As to distance and direction, rope-power transmission is practically unlimited.

(3) The amount of power which may be transmitted is practically unlimited.

(4) The cost of installing and maintaining rope drives is much lower than for belts.

(5) Transmission is more positive and steady.

(6) Rope drives are noiseless.

(7) No electrical disturbances are created.

(8) There is no measurable loss from slippage.

(9) Alinement of shafting is not necessary.

(10) Rope affords the cheapest and simplest method of distributing power to the different floors of a factory.

(11) Future additions to power may be made at small cost.

(12) Rope drives may be run outdoors, and are not affected by heat, cold, moisture, dirt, or oil.

We will admit that, if these quotations are correctly given, they should be somewhat modified to make apparent that the whole is a comparison with leather belting, and also for other reasons; for instance (9), regarding alinement, cannot possibly be a correct quotation from any reputable manufacturer's statements. Broadly speaking, the quotations as given present the points at which the advocates of rope driving endeavor to show superior merit in their article. C. W. Hunt has been justly cited as an authority on rope transmission and his opinion is as follows:

"The advantages to be found in this method of transmission of power are:

(a) Transmitting large amounts of power.

(b) Run in any direction or to any distance.

(c) Smooth and quiet running.

(d) Absence of electrical disturbance.

(e) Economy in first cost and maintenance.

(f) Absence of slip."

A comparison of the two groups of claimed advantages shows the views advanced in both to be practically identical, and it must be granted that if one is "disingenuous when not false" the other lies under the same imputation. As a matter of fact, Mr. Emerson, in his comments on the quotations first given, has not been entirely fair as to what the rope people claim to be advantages in their method of transmission.

Taking up the quotations and comments individually, (1) is dismissed as "true but unimportant." Why, in summing up *pros* and *cons* of a particular problem, should any advantage be set aside as "unimportant?" This advantage may be or may not be important. As viewed from the large engine-builder's standpoint—or any other, where short-spaced centers of bearings are of moment, or where transmission spaces are restricted—the question may be of very considerable importance.

Quotation (2) is summed up thus: "In all but the rarest cases, belts or shafts or gearing are superior to ropes." This is so plainly biased that argument seems useless. As stated before, other methods than belting should be eliminated unless the writer intends to say that some *combination using leather*, except "in the rarest cases," is better than rope. Again, the "rarest cases" means a great deal, and it is to some of them that the rope manufacturer draws particular attention. When one considers the stupendous number of power-transmission units in this country alone, although the "rare cases" constitute a very small proportion of the whole, the aggregate of them is seen to assume vast dimensions. Probably to one case where a simple arrangement of rope transmission may displace to advantage an expensive or nonefficient arrangement of belting or shafting or gears (or belting and shafting and gears), there would be hundreds of cases where the rope advocate would not for a moment urge the adoption of his system. Simply the attention of one seeking a solution of a perplexing problem in transmission is drawn to those merits conspicuous in rope, and the rope manufacturer awaits with perfect composure the outcome, when the question is to be solved intelligently. If rope is best adapted to the case it should be used; it it is not, no one is more interested to disclose the fact and advise against its use than the same manufacturer, and this is the course, as previously stated, which has been pursued by the reputable makers of transmission rope.

Quotation (3). Regarding the amount of power, it appears from Mr. Emerson's comments that he evades comparisons between leather belting and rope when large amounts of power are to be transmitted, as well he may, for it is on this line that rope has one of its most powerful arguments. It is difficult to understand why, in a candid treatment of the subject from the leather-belt standpoint, the use of rope for large power purposes should be compared with the shaft of a Cunard steamer. We may be sure, however, that in any problem of at all similar character, rope would be much more available than belting.

We are also at a loss to understand how increasing the number of ropes decreases the efficiency of a drive, and why the limit is soon reached. Really the efficiency of a drive remains practically the same regardless of the number of ropes used, and the number of ropes required will be in direct proportion to the load which is to be carried.

Quotation (4), as to the cost of installation and cost of maintenance. In making comparison between rope and leather belting with respect to cost and maintenance no attempt need be made to avoid facts. The superiority of rope in these respects does not appear until the drive is exceptional, either because of the large amount of power carried, or the distance to which the power is conveyed, or the use of the power in varying planes.

The relative merits of the two systems, as to first cost, may be very exactly obtained before installation is made, and we shall not enter upon the subject, as it is conceded "that the cost of installation may in certain cases be less."

As to the maintenance, it is difficult to regard Mr. Emerson's figures seriously. Early in the article the maintenance, with belts, is stated to be under 5 per cent. of the initial cost. This, it is to be presumed, is per annum. Later, 1 per cent. per annum is stated. The first figure is a better one than average practice reaches, the latter is preposterous.

As Mr. Emerson cites the transactions of the American Society of Mechanical Engineers for references, let us take the

"Notes on Belting," by Fred W. Taylor, one of the most carefully prepared articles which has appeared on this subject and which was presented at a meeting of the society named. Mr. Taylor's very exhaustive compilations lead him to conclude that with a load carried most economically by belting, the maintenance is not more than 14 per cent. of the first cost per annum; but that, if the belts are run with the loads of average practice, the maintenance will amount to an average of 37 per cent. of the first cost per annum. In the authoritative and intelligent discussion of this paper which followed, no criticism of these figures resulted.

The great difference between the figures as quoted from Mr. Emerson's paper and those presented by Mr. Taylor makes apparent the fact that important modification of the first should be made. It may be allowed that Mr. Taylor's deductions were not drawn from fields available to rope driving, but, in making comparisons, should not Mr. Emerson select for examples average good practice in both the belting and the rope fields?

If we accept the figure of 5 per cent. annual cost of maintenance in the belt plant, it should be clearly understood that it is an ideal figure in a plant where the belt is run under excellent conditions and receives excellent care. Now, in all fairness, the rope drive should for comparison be selected from its own ideal field. If a drive is installed with low first cost the objective, or with a shaft or shafts to be driven which are practically inaccessible by other methods of driving, or if it is one of those "rare cases" where rope is used because nothing else can be, proper allowance for these advantages should be placed in the credit column of rope driving, and maintenance alone should not be considered. This may seem to be in the line of a defense of excessive cost of maintenance in rope driving, but it is not so intended.

The intention is rather to have the matter considered fairly, and due allowance made as follows: *First*, that rope drives are many times installed where nothing else is feasible; *second*, that the care of rope is not generally in hands so experienced in its needs as is the belt, which, *thirdly*, results in overlooking the care required to keep belts in proper order, but makes very noticeable the attention demanded by the rope. In other words, the average millman is blind to the fact that his belt stretches during its life nearly as much as the rope; that it has to be respliced; that it fails under load, and so on; all of which facts come very prominently before his eyes in the case of a rope when it calls for some of his care.

To return to the cost of maintenance, as just stated, in accepting the figures of 5 per cent. of first cost in maintenance of an ideal belt drive, we will use 8 per cent. of first cost in maintenance of a parallel rope drive. That is, one run under fair conditions and installed with low cost of maintenance as an important objective.

The first cost of the leather-belt drive as a whole would be, for, say, a 500-horsepower drive, with approximately 50-foot centers, about $1270, for the rope drive about $850. Now the per cent. cost of maintenance should not be from these figures, but from the cost of the belt or rope. Let 6 per cent. be charged off each drive for fixed charges of pulleys and sheaves; this would equal, at first cost of belt pulleys of $375, $22 for the belt, and at a first cost of rope sheaves of $610, $37 for the rope. The first cost of the belt on such a drive would be approximately $895, and that of the rope some $250. The maintenance of the belt, at 5 per cent., would be $45 annually, and for the rope, at 8 per cent., $20 annually, making a total annual cost of $67 for the belt, and for the rope $57. These figures, if the distance of centers is reduced, would become more favorable to the belting, or if extended would favor the rope, but, as already mentioned, may be predetermined for any installation. The scrap value of rope is very considerably higher per cent. of the first cost of the rope than is that of leather to its first cost, being about 15 per cent., while that of leather is about 10 per cent. These figures are, of course, susceptible of wide ranges, but are relatively close averages and in favor of the belt. Attention is simply brought to the fact that the statement "maintenance costs are higher" cannot be accepted and that each particular case must be considered in itself.

Quotation (5). Overlooking the comparisons made with gearing and locomotive practice, which are hardly to be considered as pertaining to the subject of real discussion, and analyzing the statement that "one of the greatest superiorities of belting is that it is elastic and yields to varying strains, instead of breaking itself or the installation," we feel sure that there can hardly be any intention of making the statement that any relief from overload or varying strain in a belt drive comes from the elasticity of the belt. Any such relief in either a leather belt or rope drive must come from slippage, and slippage alone. We question, however, whether in any fair practice in either system, breakage of any part of the installation comes from inability to slip under load. Breakage of a rope under such conditions is almost unknown, and breakage of a belt is not common. Breakage of the shaft in either system comes from deflection, and that system which allows shortest distance centers has certainly a marked advantage. In one of the examples of rope-system failure given in the latter part of the article, a long hub pulley was doubtless replaced with a narrow-hub rope sheave, and had the bearings been relocated, no trouble would have ensued and a stiffer job would have resulted. The simple facts of the support given to a shaft by a long hub in a belt drive, and of a more concentrated load in the rope drive, are lost sight of not infrequently. The result which may come when a short-hub rope sheave is substituted for a wide-belt pulley is not understood, but the rope in some mysterious way is held accountable for failure if the same occurs.

Quotation (6). Admitting that noise in general transmission work does not count to any considerable extent one way or the other, it must be fairly conceded that while rope transmissions are practically noiseless, leather-belt ones are not, or let it be stated that rope drives are very much more quiet in operation than are leather-belt ones.

Quotation (7) In admitting the superiority of rope in its absence of electrical effects due allowance ought to be made in considering other advantages by giving it the weight it really bears in any given case. It can hardly be expected that any rope manufacturer would urge the adoption of his system for this advantage alone, however "disingenuous" he might be.

Quotation (8). The statement that "power loss in one belt is less than in one rope" is plainly incomplete and incorrect; also, the comparisons with chain and gear connections are foreign to the subject. It is well understood that the "creep" or "slipping" loss is less in ropes than in belting.

Quotation (9). The advantage claimed that in the rope drive *exact* alinement is not required does not seem to be controverted by Mr. Emerson. It is a matter of common knowledge that in belt practice shafts must approach good alinement for fair results, while in rope practice very indifferent alinement does not materially affect the drive. The statement that "bitter complaints have been made of rope drives because of careless installation" is unfortunately all too true, but why cite this fact against the rope-drive system? We may say at this point that careless and improper installation have in many instances discredited the rope system, but such installations should not be brought forth as evidence in regarding the true merit of the system itself.

Quotation (10). In discussing the feature of rope affording a cheap and simple method of reaching different floors of a factory, the article does not look at the matter broadly. In main drive problems, from motive power to different rooms of factories, large belts are almost never used on tight and loose pulleys. Such designing would be cumbersome and expensive in heavy work. The friction clutch of good design when properly installed is a valuable feature in any belt- or rope-driven mill and the value remains quite apart from its being a belt- or rope-driven one.

Again, when the statement is made that "under comparable conditions the rope

drive is not cheaper per horsepower actually transmitted," is it admitted that there are any conditions under which belt cannot be compared to rope? This does not seem in accordance with other parts of the article. The rope-driving advocate can safely assert that under "comparable conditions" his system is very much lower than the belt system in *first cost.*

Quotation (11). As to future additions, it is surely simpler to leave extra grooves in rope sheaves, where future growth is to be expected and to add the ropes as demanded by load conditions than it is to *remove* a belt in use and *replace* it with one wider or thicker, even if the extra pulley face which may be required has been anticipated.

As to Quotation (12), the article is both correct and incorrect. Ropes *are* affected by heat, by cold and by moisture, but are not rotted by oil as leather is. As a matter of fact, rope, when conditions demand, will give satisfactory service running out of doors and exposed to heat, cold and moisture, where leather or other belting would fail.

Mr. Emerson proceeds to cite cases of failure of rope drives, four in all. The first drive mentioned had the merit of working well at times, and, as no information is given as to why it worked badly at others, conclusions cannot safely be drawn. The second and third illustrations also lack details to show whether the design was incorrect or whether the rope was not adapted for the work. Certainly no conclusive reasons are given why rope driving should be condemned. It is also noted that in two of the three examples named the rope was replaced, not with a belt, but with a shaft. It would appear from the fourth example that the failure was clearly from deflection of the shaft, which, as already stated, occurs in rare cases when a rope sheave replaces a belt pulley. This is wholly a matter of design and quite outside the question of the comparative merits of the two systems.

In concluding our commentary it may be said that advocates of rope driving themselves have in many instances wrought great harm to the advancement of their cause, which is comparatively new to that of belting. At first very erroneous ideas prevailed about the construction of rope sheaves, the arrangement of drives and the proper loads to be carried. All these things have held back, probably to a large extent, that legitimate growth to which the obvious merits of rope driving entitle this system. But despite them, there has been a very large, constantly expanding growth of the rope system, and as designers and constructors grow conservative, and, instead of advising the use of rope in places where abuse is invited, use it in those cases where its good points are so conspicuous as to warrant its adoption, there is no question but that the rope system will continue to grow and will hold the position in the array of mechanical devices which it so admirably is qualified to fill.

Complimentary Smoker

The annual complimentary smoker of Phoenix Association No. 24, N. A. S. E., of New York, was held on Tuesday evening, December 8, at the German Masonic Temple building. No social session ever held by this association was more thoroughly enjoyed. Of course, as usual, the late comers had to stand, but this made no difference; everyone was there to have a good time, and they had it. There were a great entertainment and an abundance of refreshments. The next social event of Phoenix will be the banquet at the Broadway Central hotel, on February 3.

N. Y. Engineers' Protective Association

The New York Engineers' Protective Association No. 23, N. A. S. E., held its banquet and reception on Saturday evening, December 12, at Raub's café, Fulton street, New York. This was the first social event to which the ladies were invited that has been given by No. 23 for many years, and in consequence the fair sex attended in goodly numbers, making the occasion a pronounced success. Nearly two hundred guests were seated at the neatly decorated tables, at which was served a choice menu. William J. Daly officiated as toastmaster and introduced Joseph F. Carney, Thomas Trolsen, Frank M. Paulsen, W. D. Sprague and Jack Armour, who delivered toasts apropos to the occasion. Dancing followed, between the numbers of which songs were rendered by J. Lloyd Wilson, William Murray and Joseph McKenna. The committee comprised Frank M. Paulsen, Thomas Trolsen, Charles Kenney and Andrew Carlsen.

Business Items

The American Steam Pump Company, Battle Creek, Mich., is sending out a neat steel-engraved calendar for 1909.

The Ball Engine Company, of Erie, Penn., recently opened an office in the Board of Trade building, Boston, Mass., in charge of V. F. Holmes.

The Babcock & Wilcox Co. has purchased from the Rust Boiler Company its patents and plant located at Midland, Penn., and will continue the manufacture, at that point, of the Rust water-tube boiler.

An attractive booklet descriptive of the "Swartwout" gravity-closing ventilators has just been issued by the Ohio Blower Company, Cleveland, Ohio. Simple illustrations present the special features of design and construction which make this ventilator distinctive.

In addition to the sizes of Rochester lubricators which have been on the market for some time, Greene, Tweed & Co., the manufacturers, have decided to manufacture the following new sizes: Two-pint single, two-pint double, half-gallon single, half-gallon double. A new price list has been issued covering all the sizes, new and old.

The Triumph Electric Company, Cincinnati, Ohio, has made recent sales which include seven engine-type generators ranging from 75 to 200 kilowatts; also 20 motors ranging in size from 10 to 30 horsepower and aggregating 400 horsepower. These motors, including a 200-kilowatt generator, were for the Greenpoint Metallic Bed Company, Brooklyn, N. Y.

The Sight Feed Oil Pump Company, of Milwaukee, Wis., has appointed Julius A. Gerwen district representative at Duluth, Minn., with offices at 226 West Michigan street. L. G. Finley has been appointed Ohio representative with offices at 710 New England building, Cleveland, Ohio. In both of the above-mentioned offices working models of the Richardson sight-feed oil pump, as well as the pumps themselves, are on exhibition.

The Northern Water Softener Company, of Madison, Wis., has appointed D. D. Pendleton its exclusive representative for the Pittsburg district for the sale of the Bartlett systems for softening and purifying feed water. Mr. Pendleton was formerly associated with the Westinghouse company, and has had a very broad experience and a thorough knowledge of power-plant equipments. His office will be at 574 Frick building annex, Pittsburg, Penn.

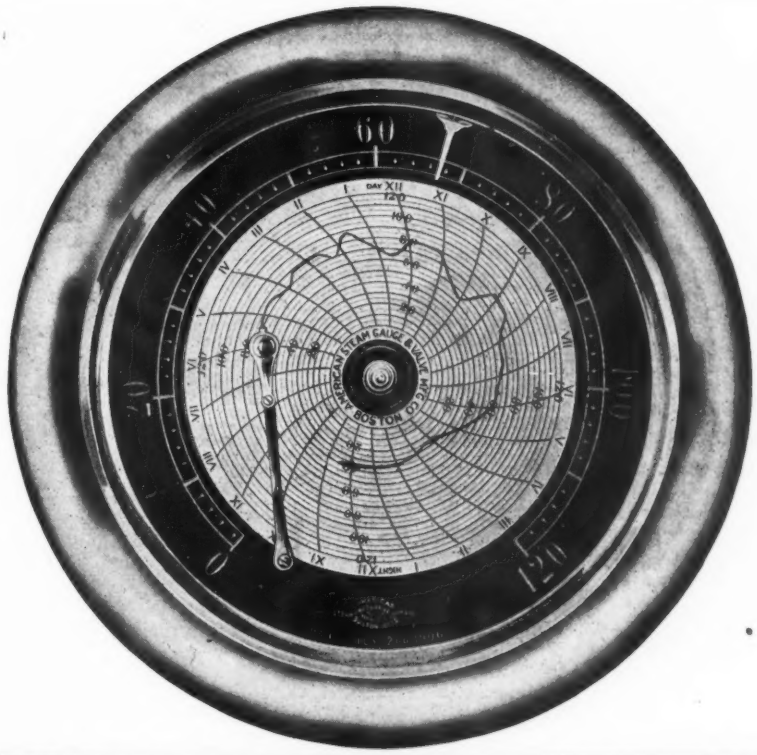
Herbert E. Stone has just become connected with the Dearborn Drug and Chemical Works, as manager of sales in the Eastern department, with headquarters at 299 Broadway, New York. Mr. Stone was formerly president of the N. A. S. E., and recently manager of the Pittsburg office of the Chapman Valve Manufacturing Company. Mr. Stone will have associated with him a corps of able assistants. The Dearborn company is to be congratulated upon the splendid addition to its selling force.

The Bruce-Meriam-Abbott Company, of Cleveland, O., reports the following recent sales of four-cylinder gas engines: One 135-horsepower to the Willard Storage Battery Company, Cleveland, O.; one 125-horsepower to the McHenry Electric Service Company, McHenry, Ill.; one 135-horsepower to the Link Ice Cream Company, Milwaukee, Wis.; two 135-horsepower to the Salamanca Light and Power Company, Salamanca, N. Y.; two 135-horsepower to the Pennsylvania Reform School, Morganza, Penn.

We have just received from the Mesta Machine Company, of Pittsburg, Penn., a 15-page pamphlet (6¼x9¼ inches) descriptive of the Helander barometric condenser, built in the United States solely by the Mesta Machine Company. While not an entirely new device, this condenser is sufficiently unfamiliar to most of our readers to warrant their sending for a copy of the pamphlet, or catalog, which is excellently printed and illustrated on embossed paper. The peculiar advantages claimed for the condenser are presented in an especially forceful and interesting manner.

The Defiance Machine Works, of Defiance, Ohio, writes to C. P. Bassett, of Charlotte, Mich., as follows: "We have had in daily use for the past four months under two of our large boilers your McNaughton patent sectional grate bars, and we find that they effected to us a saving of about 10 per cent. in fuel. They give us a more uniform fire and a better draft than we have ever received with any other style of grate bar, and another important thing is that they do not burn out, sag or get out of shape, as is the case with the old-style bars; and if they did burn out, it is very easy to replace one or more sections, and at a small expense."

The American Coal Products Company announces that it has sold its by-product coke-oven interests heretofore conducted by its subsidiary company, the United States Coke and Gas Company, to the Oberschlesische Koks-



The Primary Object

In The Manufacture Of

American Specialties

is to produce the best article of its kind and then build it at the lowest possible cost by the employment of the best skilled labor and up-to-date machinery. ¶ This accounts for the popularity of American Specialties. ¶ Cheap goods may hold the market for a time, but to maintain a business for 60 years, as we have, and enjoy the full confidence of customers, the goods must have an individual quality of their own.

One of our well-known Specialties is the AMERICAN COMBINED PRESSURE AND RECORDING GAUGE, shown above, which accurately and reliably records the exact pressure carried every minute of the day and night. ¶ Manufactured for all pressures of water, steam, gas, oil, air, etc. ¶ Let us send you Circular 3-B, which describes it.

American Steam Gauge & Valve Mfg. Co.

208-220 Camden Street, Boston, Mass.

New York, 26 Cortlandt St. Atlanta, 835 Equitable Bldg. Chicago, 7-9 So. Jefferson St.

Pittsburg
Montreal, 444 St. James Street

San Francisco, Monadnock Bldg.
Los Angeles, 213 So. Los Angeles Street

werke and Chemische Fabriken Aktien Gesellschaft, of Berlin, Germany, the transfer to take effect Jauary 1, 1909. A new company, called the German-American Coke and Gas Company, is to be formed, with headquarters at 17 Battery place, New York. Its capital is to be $2,000,000, of which $500,000 is to be paid in at once. The company will be governed by a board of directors, the officers of which will be: Chairman, W. H. Childs; president, Dr. F. Schneiwind; vice-president, A. Zindler; secretary and treasurer, F. Moser.

Help Wanted

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

WANTED—An up-to-date centrifugal pump designer. Box 74, POWER.

SALESMEN WANTED to sell a high-grade steam specialty. Must be competent. Address P. O. Box 467, Altoona, Pa.

AN ENGINEER in each town to sell the best rocking grate for steam boilers. Write Martin Grate Co., 281 Dearborn St., Chicago.

WANTED—Thoroughly competent steam specialty salesman; one that can sell high-grade goods. Address "M. M. Co.," POWER.

WANTED—By prominent engineering house on the North Pacific coast, a practical mechanical draftsman. One acquainted with power plant design preferred. Splendid opening for one who is in earnest. Address, giving full particulars as to qualifications, salary wanted, etc. Box 65, POWER.

AGENTS to sell one of the best known and widely advertised shaking grates on the market. Exclusive territory granted to anyone who can make good. Liberal commission. Perfection Grate Co., Box 1081, Springfield, Mass.

Situations Wanted

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

I WOULD LIKE TO RECOMMEND a first class fireman to anyone in Chicago needing the services of such. H. Bebb, Supt. City Fuel Co., 4012 Wentworth Ave., Chicago, Ill.

A PRACTICAL MECHANIC, 40 years old, sober, no family, familiar with the steam engine, wants situation in some first class power station to learn steam and electrical engineering. Willing to work cheap for the first year. P. O. Box 56, Nashville, Tenn.

SALESMAN — ENGINEER — Power and transmission equipment. Specialist in pumping machinery. No contract wanted until ability is fully demonstrated. Eighteen years' experience. Box 73, POWER.

POSITION WANTED as chief engineer or electrician of large plant. Expert electrician, practical machinist and steam engineer. Can operate at lowest cost, wind armatures, (all kinds), install machinery, make steam and electrical tests, use indicators. Experience on both a.c. and d.c. systems, high tension. Thirty-two years old, reliable, sober and industrious; best references; technical education. At present master mechanic of industrial plant. Box 72, POWER.

Miscellaneous

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

PATENTS secured promptly in the United States and foreign countries. Pamphlet of instructions sent free upon request. C. L. Parker, Ex-examiner, U. S. Patent Office, McGill Bldg., Washington, D. C.

DRAFTSMEN—Fit your drawing table with a parallel device, $2.50. F. G. Hobart, Beloit, Wis.

For Sale

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

FOR SALE—Direct-connected engine and generator. 100 kw.; direct current; good as new. Address Monarch Machine Works, Altoona, Pa.

FOR SALE—20x48 Wheelock engine and two 72"x18' high pressure tubular boilers in good condition cheap. Address "Engineer," Box 2, Station A, Cincinnati, Ohio.

FOR SALE—20"x42" improved Greene engine. Wheel 32"x14'. Used seven years. Also 24"x48" improved Greene engine. Wheel 42"x16.' Used eight years. Both engines in first class shape. Can be seen running. The Capewell Horse Nail Co., Hartford, Conn.

QUEEN ACME portable testing set with leather case; perfect condition; $50. V. R. Hughes, 228 Majestic Bldg., Denver, Colo.

150 HORSE-POWER tandem compound Corliss engine, in good order; 16-ft. wheel; 24-in. face. F. W. Iredell, 11 Broadway, New York.

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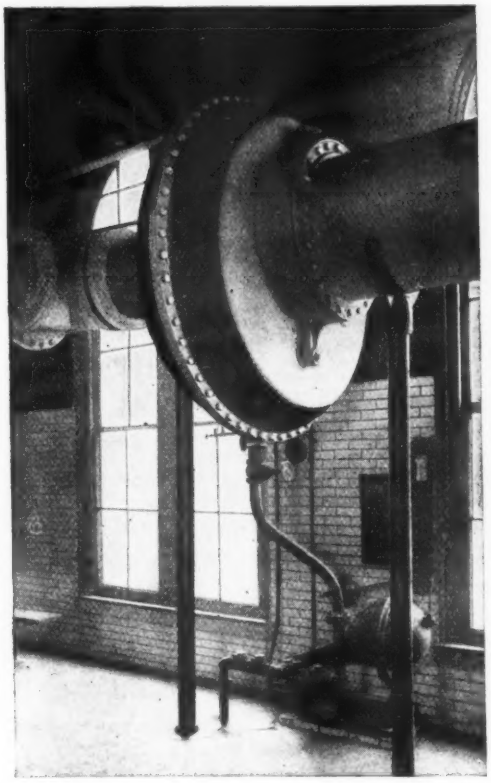
COCHRANE SEPARATORS

HOW TO REMOVE OIL FROM CONDENSED STEAM

Keep it out! That is the only satisfactory way.

Other methods that have been tried are filtration, electrical treatment, flotation and evaporation, but none of these did the trick.

Some years ago exhaustive experiments were made by J. R. Bibbins, at the plant of the Detroit Edison Company, on the removal of oil from oily drips. Pressure and heat, both combined, vacuum, aeration (hot and cold), chemicals and filtration, all were tried, but none, short of an expensive chemical treatment, was found to remove the oil satisfactorily so that the water could be used for boiler feed again.



The reason that oil cannot be filtered from condensed exhaust, is, apparently, that each particle of oil has an oxidized coating and remains in suspension in the water indefinitely without coalescing with the other particles of oil. These separate particles break down only when subjected to high heat and pressure, as in the boiler.

A careful review of all that has been done in this line only reinforces our first statement, "Keep the oil out of the condensation."

How can this be done? In the case of exhaust steam to be used in heating and drying systems, calendering rolls, etc., by a Cochrane Oil Separator. Where the steam is condensed in a surface or jet condenser, by a Cochrane Vacuum Oil Separator.

The latter is constructed much like the Cochrane Steam and Oil Separators, but is provided in addition with a device to intercept the oil that creeps along the perimeter of the exhaust pipe. This interceptor is in the form of a collar, flushed by a current of water. The efficiency of these separators is such that, when proportioned for their work and located, connected and operated in accordance with our instructions, and while the boilers are receiving ordinary care and attention as to regular cleaning, they will so thoroughly remove cylinder oil from the exhaust of condensing engines that the steam when condensed may be safely used for boiler feed and for other purposes.

Cochrane Vacuum Oil Separators are also used with evaparators and multiple effects and their condensers to prevent liquid or solid substances being carried over into the condenser or lost in the condenser water. If you have any problem of removing liquid particles from steam or gases under low pressure, we would be pleased to give you the benefit of our experience.

HARRISON SAFETY BOILER WORKS,
17TH. & CLEARFIELD STS., PHILADELPHIA, PA.

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Alcohol Distillery Apparatus

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Arch, Back Combustion Chamber

McLeod & Henry Co., Troy, N. Y.
Mayville Specialty Mfg. Co, Mayville, Wis.

Arch, Boiler

McLeod & Henry Co., Troy, N. Y.

Arch, Boiler Door

Mayville Specialty Mfg. Co., Mayville, Wis
McLeod & Henry Co., Troy, N. Y.

Asbestos Goods

Carey Co., Philip, Cincinnati, O.
Johns-Manville Co., H. W., New York.
Restein Co., Clement, Philadelphia, Pa.

Bars, Cylinder Boring

Underwood & Co., H. B., Phila., Pa.

Batteries, Storage

Westinghouse Machine Co., Pittsburg, Pa.

Belt Fasteners

Bristol Company, Waterbury, Conn.

Belting, Leather

Garlock Packing Co., Palmyra, N. Y.
N. Y. Belting & Packing Co., New York.
Reeves Pulley Co., Columbus, Ind.
Restein Co., Clement, Philadelphia, Pa.
Rhoads & Sons, J. E., Phila., Pa.
Robins New Conveyor Co., New York.
Sawyer Belting Co., New York.
Schieren Co., Chas. A., New York.
Shultz Belting Co., St. Louis, Mo.

Belting, Rubber

Diamond Rubber Co., Akron, Ohio.
Peerless Rubber Mfg. Co., New York.
Quaker City Rubber Co., Philadelphia, Pa.

Benders, Hydraulic

Watson-Stillman Co., New York.

Bending, Pipe

Harrisburg Pipe & Pipe Bending Co., Harrisburg, Pa.
National Pipe Bending Co., New Haven, Conn.
Pittsburgh Gage & Supply Co., Pittsburgh, Pa.
Whitlock Coil Pipe Co., Hartford, Conn.

Bends, Pipe

Harrisburg Pipe and Pipe Bending Co., Harrisburg, Pa.
Pittsburgh Gage and Supply Co., Pittsburgh, Pa.

Bends, Pipe—*Continued.*

Whitlock Coil Pipe Co., Hartford, Conn.

Blocks, Chain

Yale & Towne Mfg. Co., New York.

Blowers

Beggs & Co., James, New York
De Laval Steam Turbine Co., Trenton, N. J.
Green Fuel Economizer Co., Matteawan, N. Y.
McClave-Brooks Co., Scranton, Pa.
Ohio Blower Co., Cleveland, O.
Sirocco Engineering Co., New York.
Sturtevant Co., B. F., Hyde Park, Mass.
Wing Mfg. Co., L. J., New York.

Blowers, Furnace

McClave-Brooks Co., Scranton, Pa.

Boilers

Babcock & Wilcox Co., New York.
Beggs & Co., James, New York.
Berry Engineering Co., Chester, Pa
Edge Moor Iron Co., Edge Moor, Del.
Erie City Iron Wks., Erie, Pa.
Griffith & Wedge Co., Zanesville, O.
Heine Safety Boiler Co., St. Louis, Mo.
Keeler & Co., E., Williamsport, Pa.
Kewanee Boiler Co., Kewanee, Ill.
Morrin Climax Boiler Co., Brooklyn, N. Y.
Oil Well Supply Co., Oswego, N. Y.
Parker Boiler Co., Philadelphia, Pa.
Phoenix Iron Works Co., Meadville, Pa.
Robb-Mumford Boiler Co., South Framingham, Mass.
Struthers-Wells Co., Warren, Pa.
Vilter Mfg. Co., Milwaukee, Wis.
Vogt Machinery Co., Henry, Louisville, Ky.
Wetherill & Co., Robert, Chester, Pa.
Wickes Boiler Co., Saginaw, Mich.

Boilers, Second Hand

Wickes Boiler Co., Saginaw, Mich.

Books

American School of Correspondence, Chicago, Ill.
Hill Publishing Co., New York.
International Text Book Co., Scranton, Pa.
System Co., Chicago, Ill.

Boosters

Ridgway Dynamo & Engine Co., Ridgway, Pa.

Brick, Fire

C-O-Two Furnace Co., Syracuse, N. Y.
McLeod & Henry Co., Troy, N. Y.

Brick, Radial

Kellogg Co., M. W., New York.

Building Materials, Fireproof

Carey Co., Philip, Cincinnati, O.
Johns-Manville Co., H. W., New York.

Castings, Brass and Iron

Homestead Valve Mfg. Co., Pittsburg, Pa.
Illinois Malleable Iron Co., Chicago, Ill.
McClave-Brooks Co., Scranton, Pa.
Ohio Brass Co., Mansfield, O.
Pittsburgh Gage and Supply Co., Pittsburgh, Pa.

Castings, General

Neemes Bros., Troy, N. Y.

Castings, Malleable

Jeffrey Mfg. Co., Columbus, O.

Cement, Asbestos

Johns-Manville Co., H. W., New York.
N. Y. Belting & Packing Co., New York.

Cement, Boiler

Smooth-On Mfg. Co., Jersey City, N. J.

Chimneys, Reinforced Concrete

Wiederholdt Construction Co., Chicago, Ill.

Clamp, Steam Joint

McCrea & Co., James, Chicago, Ill.

Clamp, Pipe

McCrea & Co., James, Chicago, Ill.
Simplex Engineering Co., Philadelphia, Pa.

Cleaners, Mechanical Tube

Andrews Mfg. Co., Thos., Rockaway, N. J.
Buckeye Boiler Skimmer Co., Toledo, Ohio.
Chesterton Co., A. W., Boston, Mass.
Garlock Packing Co., Palmyra, N. Y.
Lagonda Mfg. Co., Springfield, O.
Liberty Mfg. Co., Pittsburg, Pa.
McCrea & Co., James, Chicago, Ill.
Pierce Co., Wm. B., Buffalo, N. Y.
Robertson & Sons, Jas. L., New York.
Stewart Heater Co., Buffalo, N. Y.

Cleaners, Steam Flue

Henry Steam Flue Cleaner Co., Ashland, Wis.

Cleaners, Turbine Boiler Tube

Lagonda Mfg. Co., Springfield, Ohio.

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Cocks, Air

Williams Valve Co., D. T., Cincinnati, Ohio.

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Lunkenheimer Co., Cincinnati, O.
Pittsburgh Gage and Supply Co., Pittsburgh, Pa.

Cocks, Gage

American Steam Gauge & Valve Mfg. Co., Boston, Mass.
Chaplin-Fulton Mfg. Co., Pittsburg, Pa.
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Ohio Brass Co., Mansfield, Ohio.
Pittsburgh Gage and Supply Co., Pittsburgh, Pa.
Reliance Gauge Column Co., Cleveland, O.

Cocks, Steam

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Schutte & Koerting Co., Philadelphia, Pa.
Williams Valve Co., D. T., Cincinnati, Ohio.

Coils, Pipe

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National Pipe Bending Co., New Haven, Conn.

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Dearborn Drug & Chemical Co., Chicago, Ill.
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Compound, Lubricating

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Ohio Injector Co., Wadsworth, O.
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Beggs & Co., James, New York.
Cooper Co., C. & G., Mt. Vernon, O.
Griffith & Wedge Co., Zanesville, O.
Minneapolis Steel & Mach'y Co., Minneapolis, Minn.
Providence Eng. Works, Providence, R. I.
Southwark Foundry and Machine Co., Philadelphia, Pa.
Vilter Mfg. Co., Milwaukee, Wis.
Wetherill & Co., Robert, Chester, Pa.
Wheeler Condenser & Engineering Co., New York.
Wisconsin Engine Co., Corliss, Wis.

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Beggs & Co., Jas., New York.
Buckeye Engine Co., Salem, Ohio.
De La Vergne Machine Co., New York.

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Foos Gas Engine Co., Springfield, O.
Minneapolis Steel & Mach'y Co., Minneapolis, Minn.
Riverside Engine Co., Oil City, Pa.
Struthers-Wells Co., Warren, Pa.
Westinghouse Machine Co., Pittsburg, Pa.
Wisconsin Engine Co., Corliss, Wis.

Engines, Hoisting

Wisconsin Engine Co., Corliss, Wis.

Engines, Oil

De La Vergne Machine Co., New York.
Du Bois Iron Works, Du Bois, Pa.

Engines, Rolling Mill

Wisconsin Engine Co., Corliss, Wis.

Engines, Rotary

Ball-Cooley Engineering Co., New York.

Engines, Steam

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Ball Engine Co., Erie, Pa.
Ball & Wood Co., Elizabethport, N. J.
Bates Machine Co., Joliet, Ill.
Beggs & Co., Jas., New York.
Buckeye Engine Co., Salem, Ohio.
Clark Bros. Co., Belmont, N. Y.
Cooper Co., C. & G., Mt. Vernon, Ohio.
Erie City Iron Wks., Erie, Pa.
Erie Mfg. & Supply Co., Erie, Pa.
Fitchburg Steam Eng. Co., Fitchburg, Mass.
Minneapolis Steel & Mach'y Co., Minneapolis, Minn.
Phoenix Iron Works Co., Meadville, Pa.
Ridgway Dynamo & Engine Co., Ridgway, Pa.
Rollins Engine Co., Nashua, N. H.
Shepherd Engineering Co., Williamsport, Pa.
Skinner Engine Co., Erie, Pa.
Southwark Foundry & Machine Co., Philadelphia, Pa.
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Westinghouse Machine Co., Pittsburg, Pa.
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Pittsburgh Gage and Supply Co., Pittsburgh, Pa.
Sturtevant Co., B. F., Hyde Park, Mass.
Whitlock Coil Pipe Co., Hartford, Conn.

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Crocker-Wheeler Co., Ampere, N. J.
Fort Wayne Electric Works, Ft. Wayne, Ind.
General Electric Co., Schenectady, N. Y.
Sturtevant Co., B. F., Hyde Park, Mass.
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Westinghouse Elec. and Mfg. Co., Pittsburg, Pa.

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Ohio Blower Co., Cleveland, O.
Sirocco Engineering Co., New York.
Sturtevant Co., B. F., Hyde Park, Mass.

Fans, Ventilating, Mine

Jeffrey Mfg. Co., Columbus, O.

Feeders, Boiler

American Boiler Economy Co., Philadelphia, Pa.
Davis Regulator Co., G. M., Chicago, Ill.
Dean Bros. Steam Pump Works, Indianapolis, Ind.
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Pittsburgh Gage & Supply Co., Pittsburg, Pa.

Filters, Feed Water

Beggs & Co., James, New York.
Eclipse Feed Water Heater and Purifier Co., Appleton, Wis.
Keystone Chemical Co., Philadelphia, Pa.

Filters, Oil

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Burt Mfg. Co., Akron, O.
Houghton & Co., E. F., Philadelphia, Pa.
Liberty Mfg. Co., Pittsburg, Pa.
Pittsburgh Gage & Supply Co., Pittsburg, Pa.
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Filtration and Softening Plants

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Harrison Safety Boiler Wks., Philadelphia, Pa.

Fittings, Ammonia

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Triumph Ice Machine Co., Cincinnati, O.
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Fittings, Boiler

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Fittings, Flange

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Pittsburgh Valve & Fittings Co., Barberton, O.
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Watson-Stillman Co., New York.

Fittings, Pipe

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Flanges, Welded

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Gages, Steam

American Steam Gauge and Valve Mfg. Co., Boston, Mass.

Gages, Vacuum

American Steam Gauge and Valve Mfg. Co., Boston, Mass.
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Schutte & Koerting Co., Philadelphia, Pa.
Standard Gauge Mfg. Co., Foxboro, Mass.

Gages, Water

American Steam Gauge & Valve Mfg. Co., Boston, Mass.
Huyette Co., Paul B., Philadelphia, Pa.
Lunkenheimer Co., Cincinnati, Ohio.
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Gas Producers

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Westinghouse Mach. Co., Pittsburg, Pa.

Gaskets, Copper

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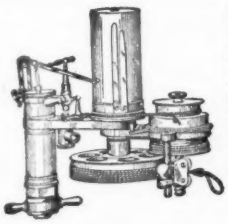

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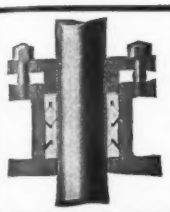

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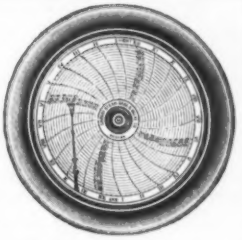

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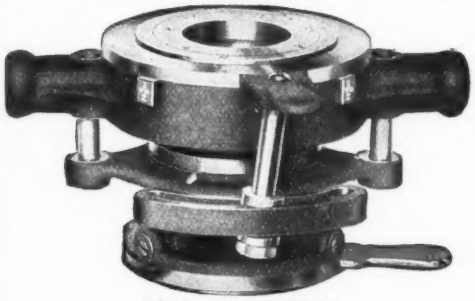

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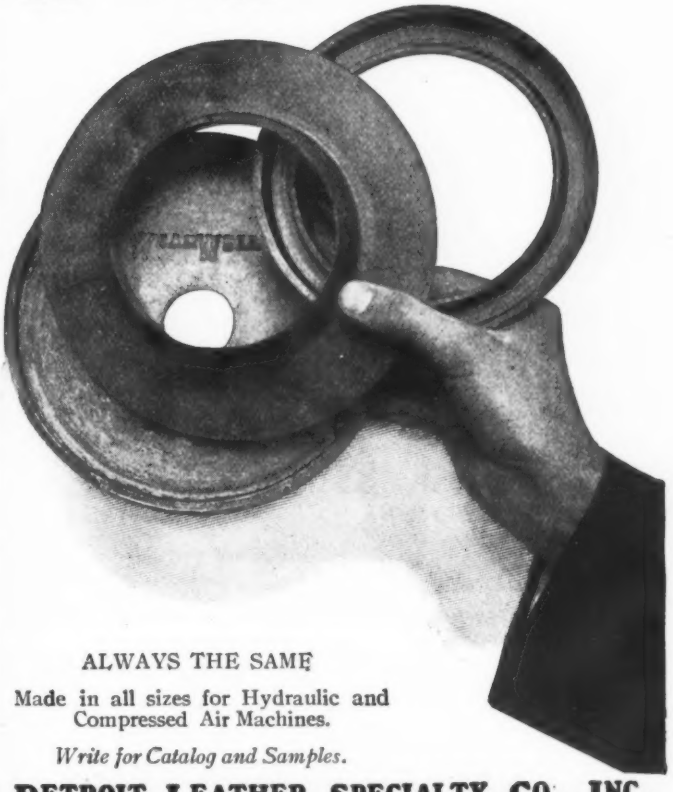

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Westinghouse Electric & Mfg. Co., Pittsburg, Pa.

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Ridgway Dynamo & Engine Co., Ridgway, Pa.

Governors, Pump

American Boiler Economy Co., Philadelphia, Pa.
Berry Engineering Co., Chester, Pa.
Chaplin-Fulton Mfg. Co., Pittsburg, Pa.
Dean Bros. Steam Pump Works, Indianapolis, Ind.
d'Este Co., Julian, Boston, Mass.
Fisher Governor Co., Marshalltown, Iowa.
Gardner, Governor & Separator Co. Quincy, Ill.
Strong, Carlisle & Hammond Co., Cleveland, O.

Graphite

Carpenter & Co., W. D., New York.
Dixon Crucible Co., Jos., Jersey City, N. J.
Garlock Packing Co., Palmyra, N. Y.

Grate Bars

Erie City Iron Wks., Erie, Pa.
Salamander Grate Bar Co., New York.

Grates, Chain

Babcock & Wilcox Co., New York.
Green Engineering Co., Chicago, Ill.

Grates, Clinker Cutting

Neemes Bros., Troy, N. Y.

Grates, Rocking

Salamander Grate Bar Co., New York.

Grates, Shaking and Dumping

McClave-Brooks Co., Scranton, Pa.
Martin Grate Co., Chicago, Ill.
Neemes Bros., Troy, N. Y.
Salamander Grate Bar Co., New York.

Grates, Stationary

Neemes Bros., Troy, N. Y.
Salamander Grate Bar Co., New York.

Grease

Carpenter & Co., W. D., New York
Cook's Sons, Adam, New York.
Harris Oil Co., A. W., Providence, R. I.
Houghton & Co., E. F., Philadelphia, Pa.
Keystone Lubricating Co., Philadelphia, Pa.

Grease Cups

See Oil and Grease Cups.

Grease Extractors

Beggs & Co., Jas., New York.

Heaters and Purifiers, Feed Water

Alberger Condenser Co., New York.
Baragwanath & Son, Wm., Chicago, Ill.
Bates Machine Co., Joliet, Ill.
Buckeye Boiler Skimmer Co., Toledo, Ohio.
Eclipse Feed Water Heater and Purifier Co., Appleton, Wis
Erie City Iron Wks., Erie, Pa.
Erie Mfg. & Supply Co., Erie, Pa.
Griscom-Spencer Co., New York.

Heaters and Purifiers, Feed Water—*Continued.*

Harrison Safety Boiler Wks., Philadelphia, Pa.
Harrisburg Pipe and Pipe Bending Co., Harrisburg, Pa.
Hoppes Mfg. Co., Springfield, O.
Kewanee Boiler Co., Kewanee, Ill
Loew Mfg. Co., Cleveland, O.
National Pipe Bending Co., New Haven, Conn.
Patterson & Co., F. L., New York.
Schutte & Koerting Co., Philadelphia, Pa.
Stewart Heater Co., Buffalo, N. Y.
Webster & Co., Warren, Camden, N. J.
Wheeler Condenser and Engineering Co., New York.
Whitlock Coil Pipe Co., Hartford, Conn.
Wickes Boiler Co., Saginaw, Mich.

Heating, District Steam

American District Steam Co., Lockport, N. Y.

Heating and Ventilating Apparatus

Green Fuel Economizer Co., Matteawan, N. Y.
Ohio Blower Co., Cleveland, O.
Sturtevant Co., B. F., Hyde Park, Mass.
Webster & Co., Warren, Camden, N. J.

Hoists, Electric

Yale & Towne Mfg. Co., New York.

Hoists, Hand

Yale & Towne Mfg. Co., New York.

Hose

Diamond Rubber Co., Akron, Ohio.
Garlock Packing Co., Palmyra, N. Y.
Lagonda Mfg. Co., Springfield, Mass.
Mercer Rubber Co., Hamilton Square, N. J.
N. Y. Belting & Packing Co., New York.
Peerless Rubber Mfg. Co., New York.
Pennsylvania Flexible Metallic Tubing Co., Philadelphia, Pa.
Quaker City Rubber Co., Philadelphia, Pa.
Restein Co., Clement, Philadelphia, Pa.

Indicators, Ammonia

American Steam Gauge and Valve Mfg. Co., Boston, Mass.
Schaeffer & Budenberg Mfg. Co., Brooklyn, N. Y.

Indicators, Gas Engine

American Steam Gauge and Valve Mfg. Co., Boston, Mass.
Schaeffer & Budenberg Mfg. Co., Brooklyn, N. Y.

Indicators, Hydraulic

American Steam Gauge and Valve Mfg. Co., Boston, Mass.

Indicators, Speed

American Steam Gauge and Valve Mfg. Co., Boston, Mass.
Schaeffer & Budenberg Mfg. Co., Brooklyn, N. Y.
Starrett Co., L. S., Athol, Mass.

Indicators, Steam Engine

American Steam Gauge and Valve Mfg. Co., Boston, Mass.
Crosby Steam Gage & Valve Co., Boston, Mass.
Lippincott Steam Specialty and Supply Co., Newark, N. J.
Robertson & Sons, James L., New York.
Schaeffer & Budenberg Mfg. Co., Brooklyn, N. Y.
Trill Indicator Co., Corry, Pa.

Injectors

American Injector Co., Detroit, Mich.
Lunkenheimer Co., Cincinnati, O.
Ohio Injector Co., Wadsworth, O.
Penberthy Injector Co., Detroit, Mich.
Randle Mach. Co., Cincinnati, O.
Schutte & Koerting Co., Philadelphia, Pa.

Insulation

Johns-Manville Co., H. W., New York.

Jacks, Hydraulic

Watson-Stillman Co., New York.

Joints, Pipe

Pittsburgh Valve & Fittings Co., Barberton, O.
Pittsburgh Valve, Foundry & Construction Co., Pittsburg, Pa.
Whitlock Coil Pipe Co., Hartford, Conn.

Kiln Equipment

Schutte & Koerting Co., Philadelphia, Pa.
Sturtevant Co., B. F., Hyde Park, Mass.

Lacing, Belt

Bristol Co., Waterbury, Conn.

Lamps, Arc and Incandescent

Fort Wayne Electric Works, Ft. Wayne, Ind.
General Electric Co., Schenectady, N. Y.
Westinghouse Electric and Mfg. Co., Pittsburg, Pa.

Lathes, Foot Power

Barnes & Co., W. F. and John, Rockford, Ill.

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Yale & Towne Mfg. Co., New York.

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Dearborn Drug & Chemical Wks., Chicago, Ill.
Dixon Crucible Co., Jos., Jersey City, N. J.
Harris Oil Co., A. W., Providence, R. I.
Keystone Lubricating Co., Philadelphia, Pa.

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American Injector Co., Detroit, Mich.
Cook's Sons, Adam, New York.
Chesterton Co., A. W., Boston, Mass.
Detroit Lubricator Co., Detroit, Mich.
Greene, Tweed & Co., New York.
Griscom Spencer Co., New York.
Lunkenheimer Co., Cincinnati, O.
Ohio Lubricator Co., Wadsworth, O.
Philadelphia Lubricator & Mfg. Co., Philadelphia, Pa.
Williams Valve Co., D. T., Cincinnati, Ohio.
See Oiling Systems.

Machinery, Coal and Ash Handling

Jeffrey Mfg. Co., Columbus, O.

Machinery, Conveying

Jeffrey Mfg. Co., Columbus, O.

Machinery, Crank Pin Turning

Underwood & Co., H. B., Philadelphia, Pa.

Machinery, Ice and Refrigerating

De La Vergne Machine Co., New York.
Triumph Ice Machine Co., Cincinnati, Ohio.
Vilter Mfg. Co., Milwaukee, Wis.
York Mfg. Co., York, Pa.

Machinery, Mining

Wisconsin Engine Co., Corliss, Wis.

Machinery, Second Hand

Randle Mach. Co., Cincinnati, O.

Machines, Blue Printing

Buckeye Engine Co., Salem, Ohio.

Machines, Milling, Portable

Underwood & Co., H. B., Philadelphia, Pa.

Machines, Pipe Cutting and Threading

Bignall & Keeler Mfg. Co., Edwardsville, Ill.
Curtis & Curtis Co., Bridgeport, Conn.
Loew Mfg. Co., Cleveland, O.
Oster Mfg. Co., Cleveland, O.
Toledo Pipe Threading Machine Co., Toledo, O.
Trimont Mfg. Co., Roxbury, Mass.
Williams & Co., J. H., Brooklyn, N. Y.

Machines, Reseating, Flue Cap

Lagonda Mfg. Co., Springfield, O.
Liberty Mfg. Co., Pittsburg, Pa.

Machines, Reseating, Valve

Leavitt Mach. Co., Orange, Mass.

Mats and Matting

Mercer Rubber Co., Hamilton Square, N. J.

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Allan & Son, A., New York.
Reeves Pulley Co., Columbus, Ind.
St. John, G. C., New York.

Metal, Babbit

Allan & Son, A., New York.

Motors, Electric

American Engine Co., Bound Brook, N. J.
Clark Bros. Co., Belmont, N. Y.
Crocker-Wheeler Co., Ampere, N. J.
Fort Wayne Electric Works, Ft. Wayne, Ind.
General Electric Co., Schenectady, N. Y.
Northern Electrical Mfg. Co., Madison, Wis.
Ridgway Dynamo & Engine Co., Ridgway, Pa.
Sturtevant Co., B. F., Hyde Park, Mass.
Triumph Electric Co., Cincinnati, O.
Westinghouse Electric and Mfg. Co., Pittsburg, Pa.

Motors, Water

Lagonda Mfg. Co., Springfield, Ohio.

Oil Burners

Hammel Oil Burner Co., Los Angeles, Cal.

Oil and Grease Cups

American Injector Co., Detroit, Mich.
Cook's Sons, Adam, New York.
Greene, Tweed & Co., New York.
Keystone Lubricating Co., Philadelphia, Pa.
Lunkenheimer Co., Cincinnati, O.
Philadelphia Lubricator & Mfg. Co., Philadelphia, Pa.

Oilers

American Injector Co., Detroit, Mich.
Lunkenheimer Co., Cincinnati, O.
Pittsburgh Gage & Supply Co., Pittsburg, Pa.

Oiling Systems

Burt Mfg. Co., Akron, O.
Cook's Sons, Adam, New York.
Harris Oil Co., A. W., Providence, R. I.
Pittsburgh Gage & Supply Co., Pittsburgh, Pa
Standard Gauge Mfg. Co., Foxboro, Mass.
Williams Valve Co., D. T., Cincinnati, Ohio.

Oil Reservoirs

Burt Mfg. Co., Akron, Ohio.

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Cook's Sons, Adam, New York.
Dearborn Drug & Chemical Wks., Chicago, Ill.
Houghton & Co., E. F., Philadelphia, Pa.

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Restein Co., Clement, Philadelphia, Pa.

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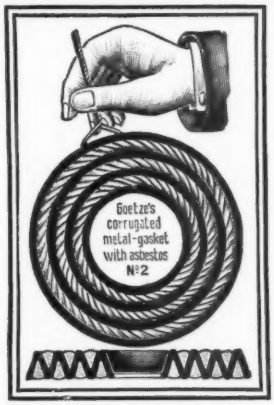

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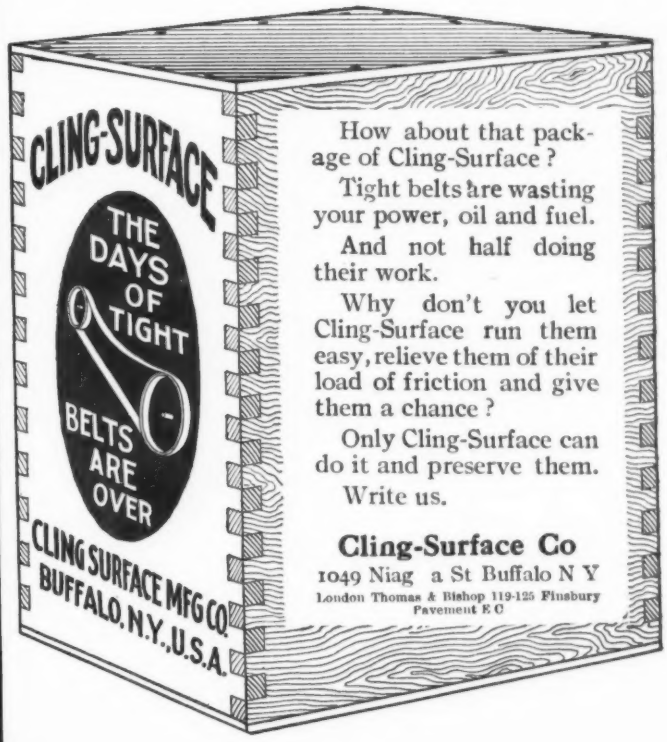

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Power Specialty Co., New York.
Restein Co., Clement, Phila., Pa.
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Canfield Mfg. Co., Philadelphia, Pa.
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Eureka Packing Co., New York.
Garlock Packing Co., Palmyra, N. Y.
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Packing, Rod
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Pipe
See Tubing

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Chesterton Co., A. W., Boston, Mass.
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Watson-Stillman Co., New York.

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Hill Publishing Co., New York.
International Correspondence Schools, Scranton, Pa.
System Co., Chicago, Ill.

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Houghton & Co., E. F., Philadelphia, Pa.
Myers & Bro., F. E., Ashland, O.
Reeves Pulley Co., Columbus, Ind.
Saginaw Mfg. Co., Saginaw, Mich.

Pulleys, Clutch
Williams Fdry. & Mach. Co., Akron, Ohio.

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Lammert & Mann, Chicago, Ill.
Schutte & Koerting Co., Philadelphia, Pa.
Warren Steam Pump Co., Warren, Mass.
Wheeler Condenser & Engineering Co., New York.

Pumps, Boiler Feed
Baragwanath & Son, Wm., Chicago, Ill.
Kewanee Boiler Co., Kewanee, Ill.
Schutte & Koerting Co., Philadelphia, Pa.
Warren Steam Pump Co., Warren, Mass.
Worthington, Henry R., New York.

Pumps, Centrifugal
Wheeler Condenser & Engineering Co., New York.
Worthington, Henry R., New York.

Pumps, Compound
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Pumps, Deep Well
Du Bois Iron Works, Du Bois, Pa.
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Warren Steam Pump Co., Warren, Mass.

Pumps, Electric
Du Bois Iron Works, Du Bois, Pa.

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Pumps, Mining
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Pumps, Oil
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Burt Mfg. Co., Akron, Ohio.
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Du Bois Iron Wks., Du Bois, Pa.
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Pittsburgh Gage & Supply Co., Pittsburg, Pa.

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Patterson & Co., F. L., New York.

Pumps, Pressure
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Pumps, Steam
Blake Mfg. Co., Geo. F., New York.
De Laval Steam Turbine Co., Trenton, N. J.
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Epping-Carpenter Co., Pittsburg, Pa.
Gardner Gov. & Sep. Co., Quincy, Ill.
Lammert & Mann, Chicago, Ill.

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Myers & Bro., F. E., Ashland, O.
Stewart Heater Co., Buffalo, N. Y.
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Wheeler Cond. & Eng. Co., New York.

Pumps, Triplex
Deming Co., Salem, O.

Pumps, Turbine
Alberger Condenser Co., New York.
Worthington, Henry R., New York.

Pumps, Vacuum
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Punches, Hydraulic
Watson-Stillman Co., New York.

Purifiers, Steam
Hoppes Mfg. Co., Springfield, O.

Purifiers, Water
See Heaters and Purifiers.

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Standard Gauge Mfg. Co., Foxboro, Mass.

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Hoppes Mfg. Co., Springfield, O.
Liberty Mfg. Co., Pittsburg, Pa.
Ohio Blower Co., Cleveland, O.
Patterson & Co., F. L., New York.
Pittsburgh Gage & Supply Co., Pittsburgh, Pa.
Robertson & Sons, Jas. L., New York.
Watson & McDaniel Co., New York.
Webster & Co., Warren, Camden, N. J.
Wheeler Condenser & Engineering Co., New York.
Whitlock Coil Pipe Co., Hartford, Conn.
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Carey Co., Philip, Cincinnati, Ohio.
Johns-Manville Co., H. W., New York.

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Buckeye Boiler Skimmer Co., Toledo, Ohio.

Skylights
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Climax Smoke Preventer Co., Boston, Mass.

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American Steam Gauge and Valve Mfg. Co., Boston, Mass.
Baragwanath & Son, Wm., Chicago, Ill.
Davis Regulator Co., G. M., Chicago, Ill.
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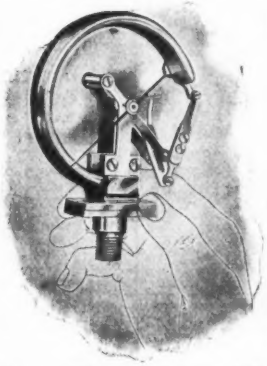

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Patterson & Co., F. L., New York.
Penberthy Injector Co., Detroit, Mich.
Pittsburgh Gage & Supply Co., Pittsburg, Pa.
Reliance Gauge Column Co., Cleveland, O.
Restein Co., Clement, Philadelphia, Pa.
Ridgway Dynamo & Engine Co., Ridgway, Pa.
Robertson & Sons, Jas. L., New York.
Schaeffer & Budenberg Mfg. Co., Brooklyn, N. Y.
Stephenson Mfg. Co., Albany, N. Y.
Strong, Carlisle & Hammond Co., Cleveland, O.
Watson & McDaniel Co., Phila., Pa.
Webster & Co., Warren, Camden, N. J.
Williams Valve Co., D. T., Cincinnati, Ohio.

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Bignall & Keeler Mfg. Co., Edwardsville, Ill.
Curtis & Curtis Co., Bridgeport, Conn.
Loew Mfg. Co., Cleveland, O.
Oster Mfg. Co., Cleveland, O.
Toledo Pipe Threading Machine Co., Toledo, O.

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McClave-Brooks Co., Scranton, Pa.
Westinghouse Mach. Co., Pittsburg, Pa.
Wilkinson Mfg. Co., Bridgeport, Pa.

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Babcock & Wilcox Co., New York.
Green Engineering Co., Chicago, Ill.
McClave-Brooks Co., Scranton, Pa.
Westinghouse Machine Co., Pittsburg, Pa.
Wilkinson Mfg. Co., Bridgeport, Conn.

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Dean Bros. Steam Pump Works, Indianapolis, Ind.
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Lunkenheimer Co., Cincinnati, O.
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Parker Boiler Co., Phila., Pa.
Power Specialty Co., New York.
Providence Eng. Wks., Providence, R. I.
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General Electric Co., Schenectady, N. Y.
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Ohio Brass Co., Mansfield, Ohio.

Supplies, Engineers'
Mound Tool & Scraper Co., St. Louis, Mo.

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American Steam Gauge and Valve Mfg. Co., Boston, Mass.
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Erie Mfg. & Supply Co., Erie, Pa.
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Schaeffer & Budenberg Mfg. Co., Brooklyn, N. Y.
Schutte & Koerting Co., Philadelphia, Pa.
Strong, Carlisle & Hammond Co., Cleveland, Ohio.

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General Electric Co., Schenectady, N. Y.
Westinghouse Electric & Mfg. Co., Pittsburg, Pa.

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Pittsburgh Gage & Supply Co., Pittsburg, Pa.

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McCrea & Co., James, Chicago, Ill.
Morehead Mfg. Co., Detroit, Mich.
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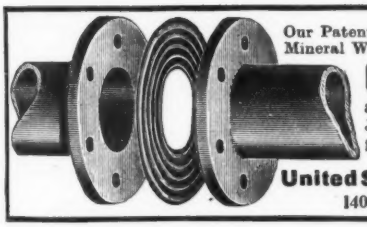

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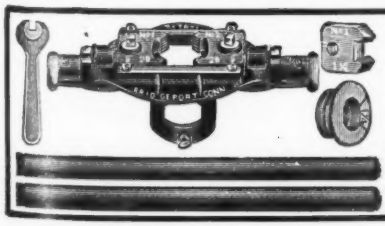

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Style 1

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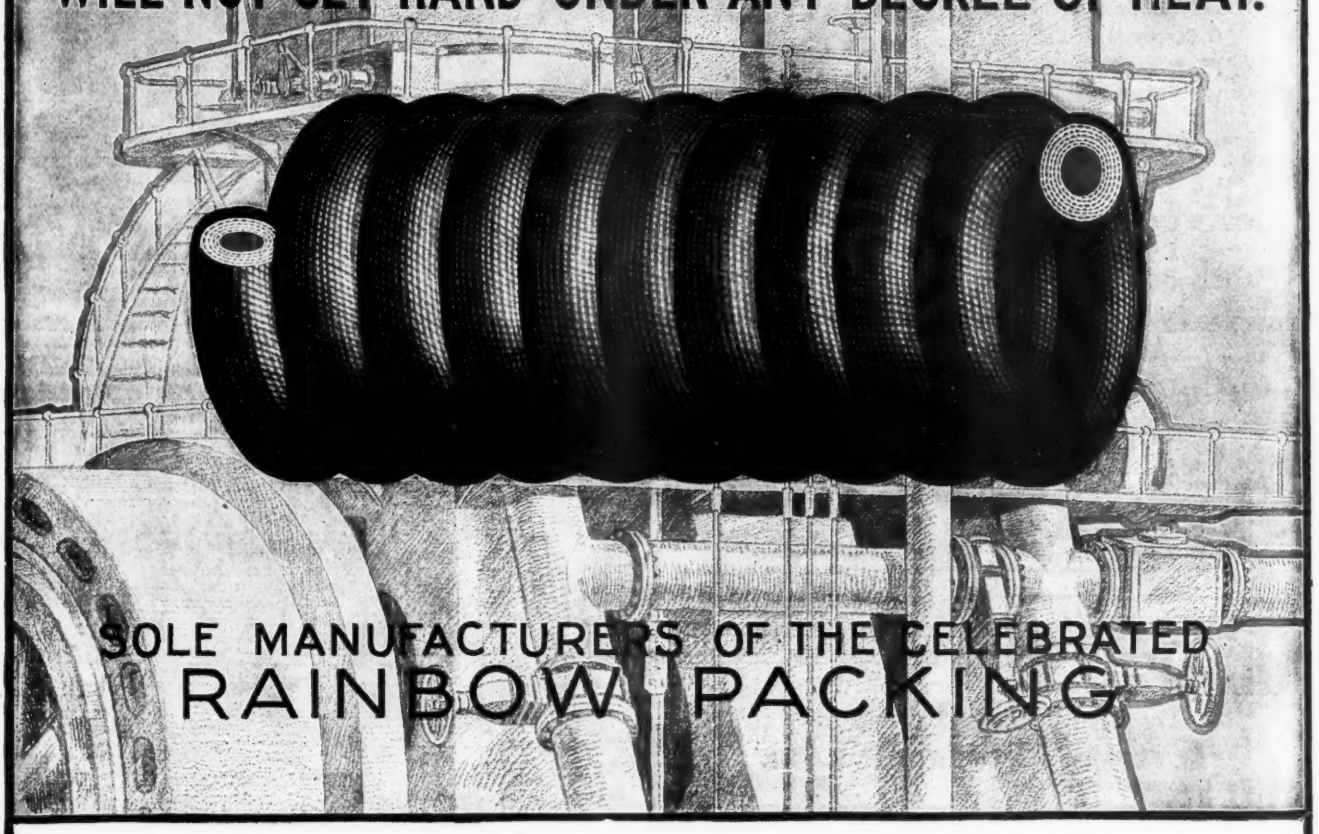

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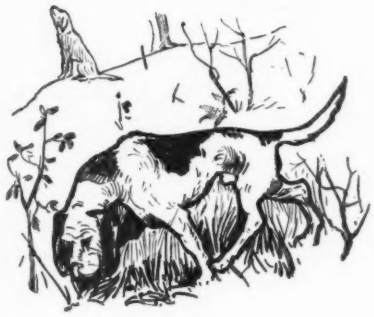

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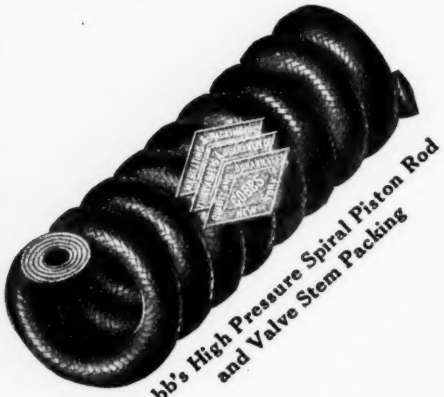

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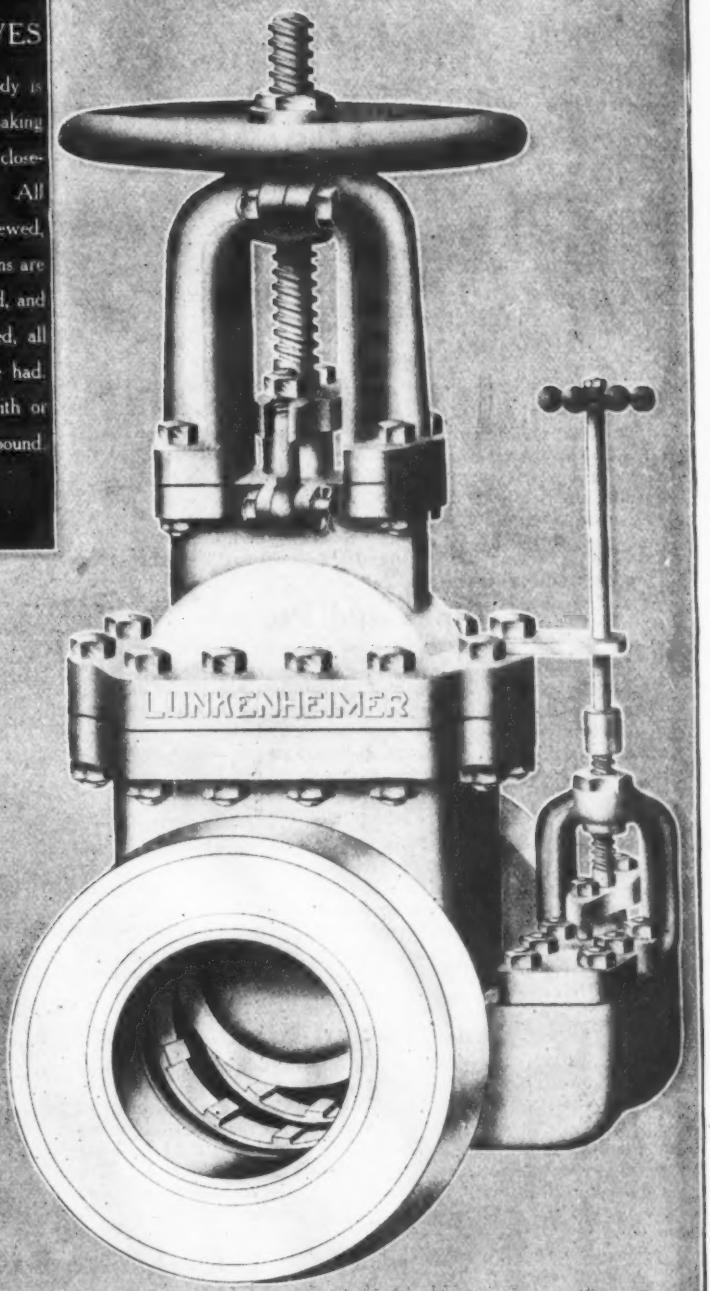
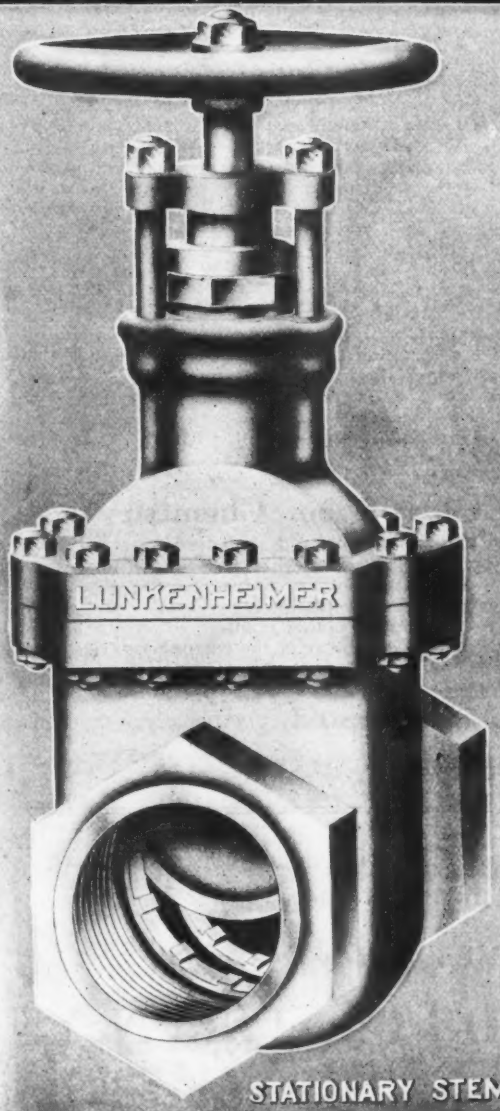


OUTSIDE SCREW AND YOKE WITH BY-PASS



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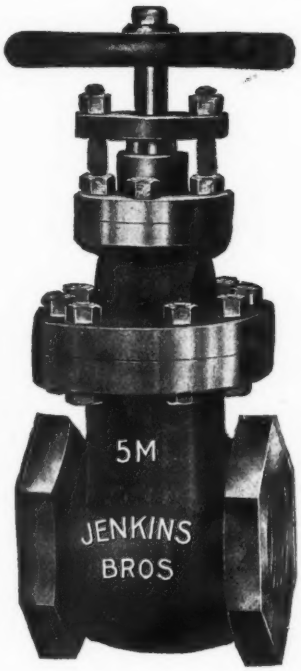

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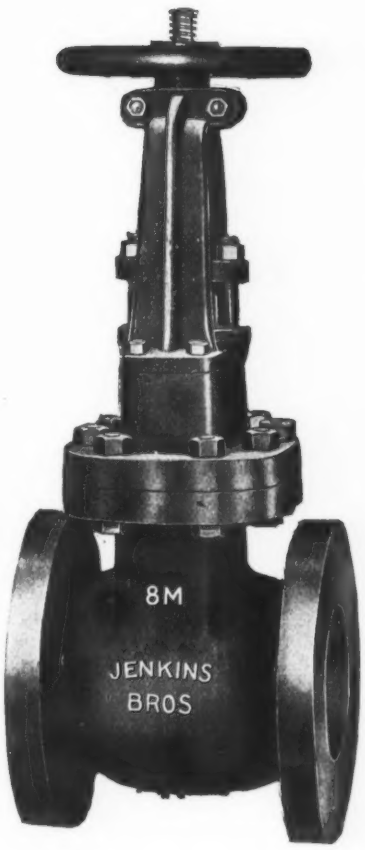
For medium heavy pressure steam and water service of all kinds these Jenkins Bros. Medium Pressure Gate Valves are the most satisfactory ever made for the purpose. They are intended for 150 lbs. working pressure but each one is carefully tested to 500 lbs. hydraulic pressure, before leaving our plant.



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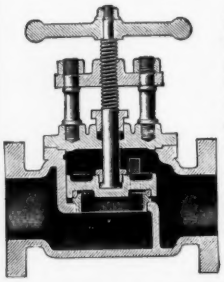

have the seat on the outside of an inverted cone where the direct current of steam does not come in contact with it, preventing all chance of wire drawing. Other features in catalog.

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Is as necessary in the boiler room as the clock is in the office. The Ashton Improved Steam Gage registers the boiler pressure far more accurately than the average clock keeps time.

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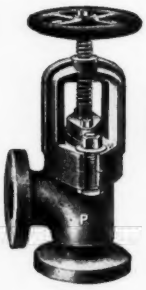

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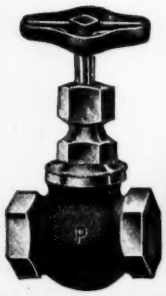

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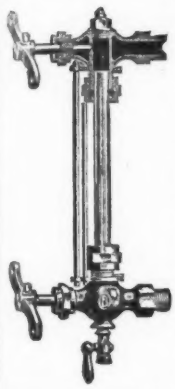

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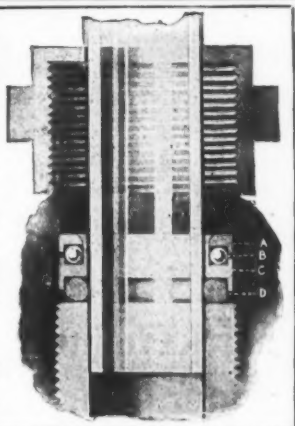

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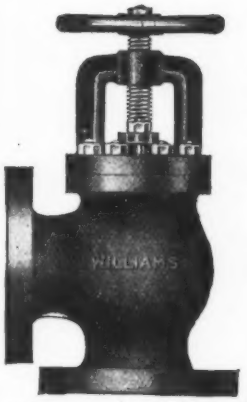

IF the valves in your plant are not giving satisfactory service, why don't you install valves you can depend upon? Williams are the valves you should use. They are the reliable kind — the kind that when once placed in operation, will never give trouble ever afterward. As they do not require constant repairs and attention, they are worthy of the consideration of every engineer who wants greater valve economy.

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are intended for the most exacting service. Correctly designed, well proportioned, they are much heavier than other valves and are strictly guaranteed for 175 pounds working pressure. If you want to repack them under pressure, you can do so without any danger of being burnt or scalded. If the seat should ever become worn, it is easily reground and a new bearing secured without any extra cost.

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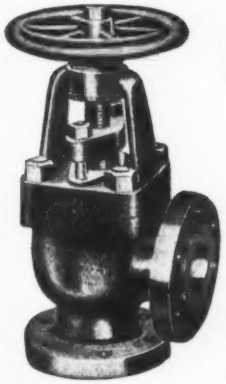

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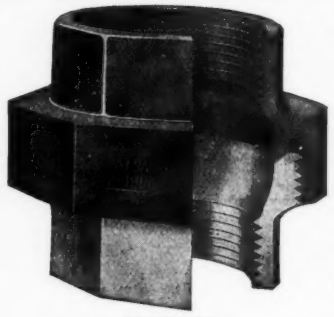

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It's easy to connect and disconnect "Kewanee" Unions. They won't stick or rust and need no pounding because they have brass to iron thread connection and cannot rust together. The ball-joint seat makes the "Kewanee" absolutely tight without a gasket. Ask for information.

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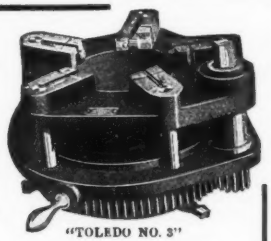
is all that this machine requires to thread pipe from 4½" to 8" inclusive.



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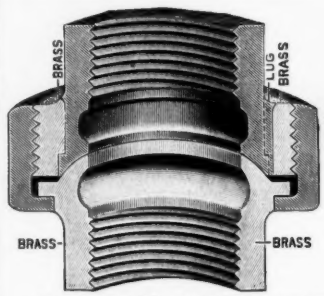

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Do you ever have Motor Troubles?

When they do come it's a case of hustle to trace and correct the difficulty.

Often it's a case of long labor, a shut-down on the line, time wasted and work delayed.

What every engineer needs is some way of knowing how to quickly and surely trace the source of motor troubles and correct them.

We provide it for you in a new book by E. B. Raymond, Electrical Engineer of the General Electric Company.

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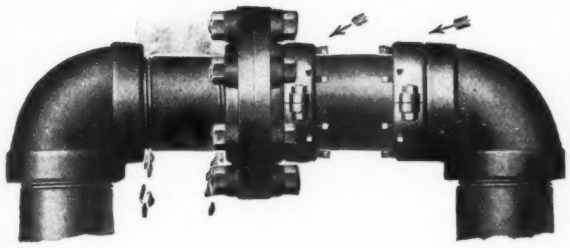

THE SIMPLEX PIPE CLAMP will stop any leak in your plant permanently, securely and inexpensively. You should have one on 30 days' free trial. It will make good on a bad leak.

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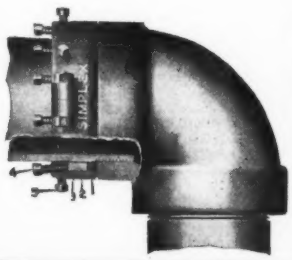

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The Drip Cock of the U. S. Automatic prevents freezing; prevents overheating of suction pipe; prevents jet "liming up" when steam or check valve leaks; *prevents destruction of hose on traction engines* by heat from hot water, and has many other exclusive advantages fully explained in our

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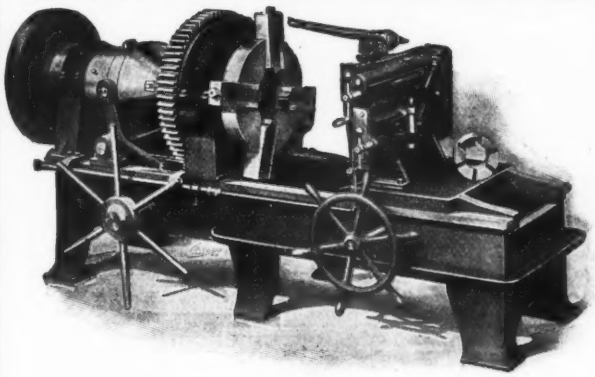

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Cannot be beat for raising water from deep wells, mines or pits, filling or emptying tanks, raising or transferring liquids, etc.

Note the straight pipe connections making it desirable for elevating water from deep driven or drilled wells.

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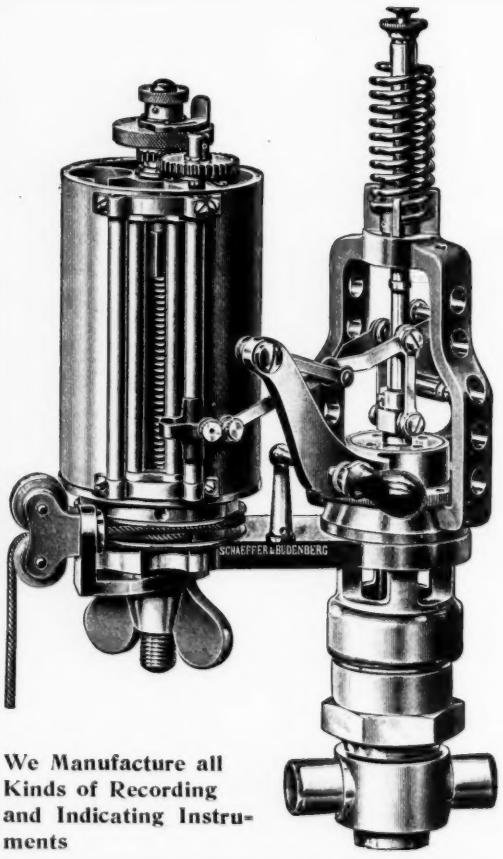

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For Steam and Gas Engines

The Improved

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With Outside Spring and Continuous Diagram Pattern 1906

Takes a strip of paper 6 feet long.

The roll can be changed without removing the indicator from the engine.

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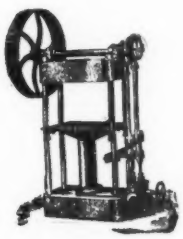

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Makes steel smoke stacks last indefinitely. It protects them from the destructive elements which soon eat out and rot unprotected stacks.

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VITRIBESTOS is far superior to brick for lining stacks, because it weighs but 1-10 as much and only obstructs 11% of the opening on a 6 foot stack, while brick takes up 26 per cent. of the space. Consequently, VITRIBESTOS does not make necessary such expensive foundations to support it, and insures much better draft.

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836

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Carey's Coverings are absolutely fire-proof, and being thorough non-conductors of heat they confine the heat to the pipes, effectually preventing its loss through radiation, thus saving in fuel an amount sufficient to *more* than offset the cost of their application.

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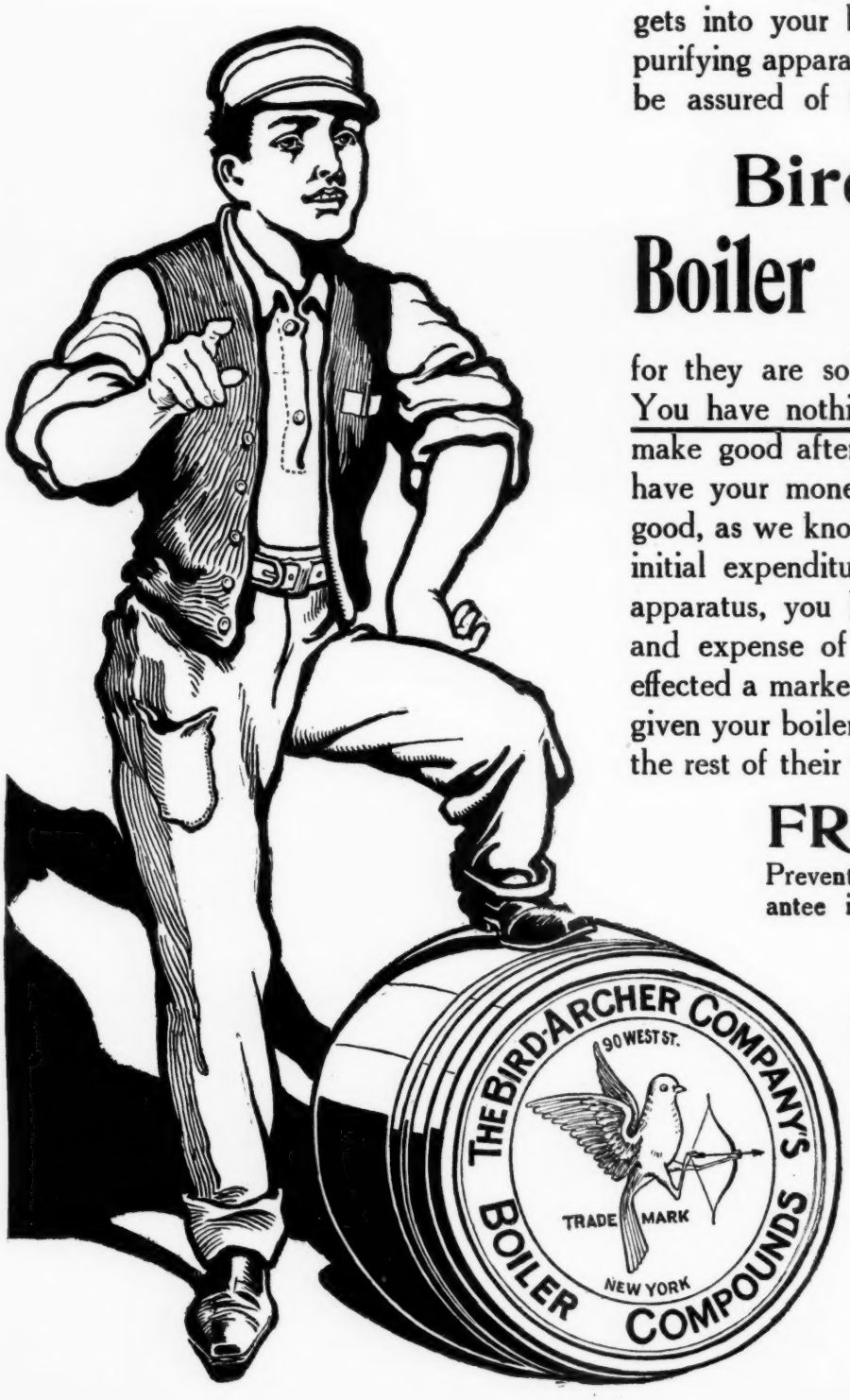

If your feed water is bad, if oil gets into your boilers, don't buy expensive purifying apparatus on a gamble. You can be assured of the desired result by using

Bird=Archer Boiler Compounds,

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FREE! Our book: "Boiler Troubles and Their Prevention" explains this and our guarantee in detail. Write for it today.

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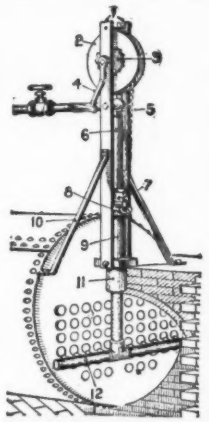
CUT YOUR FUEL BILLS

by keeping your boiler flues free from soot. The Henry Steam Flue Cleaner is guaranteed to do the business while the boiler is in commission. It is operated by simply turning a crank and opening a valve. Descriptive matter? Live agents wanted.



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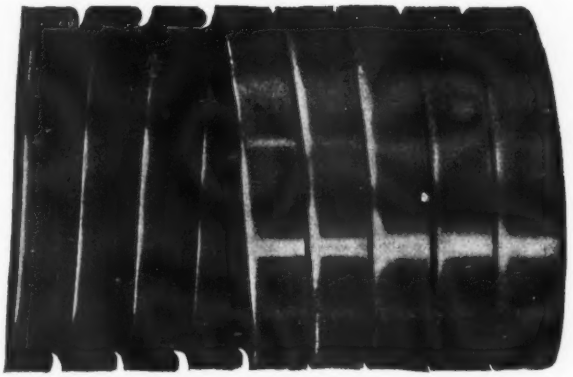

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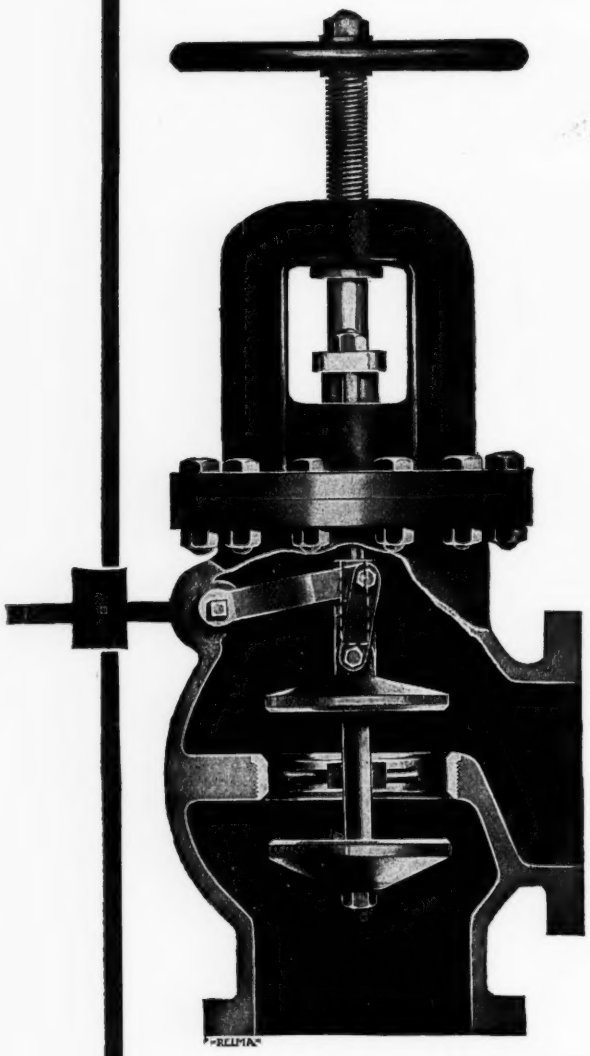
DON'T RISK FATAL STEAM PIPE EXPLOSIONS!!!



Two men were recently scalded to death and one seriously injured by a steam pipe explosion at one of the large manufacturing plants, Workington, Eng. It's the old story, engineer capable and efficient, but somebody not familiar with the sudden action of steam turned it on too quick, when "cutting in" a boiler. You can't protect your men and your plant too carefully. Nor should you run risks which can be prevented at practically no extra expense.

Accidents common to ordinary valves are impossible with the

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for the Lagonda Valve acts automatically and gradually as pressure on one side varies from that on the other.

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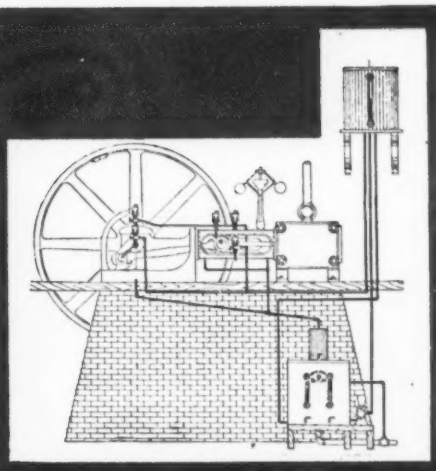

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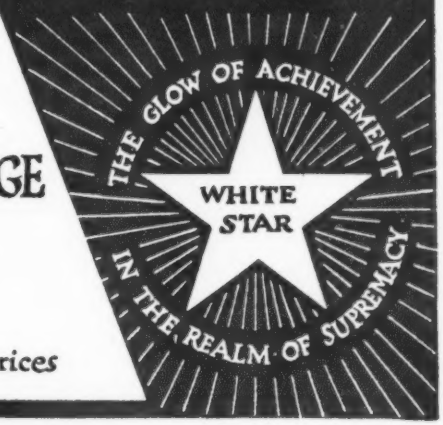

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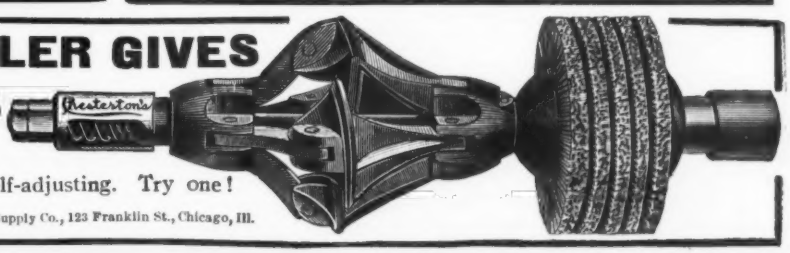

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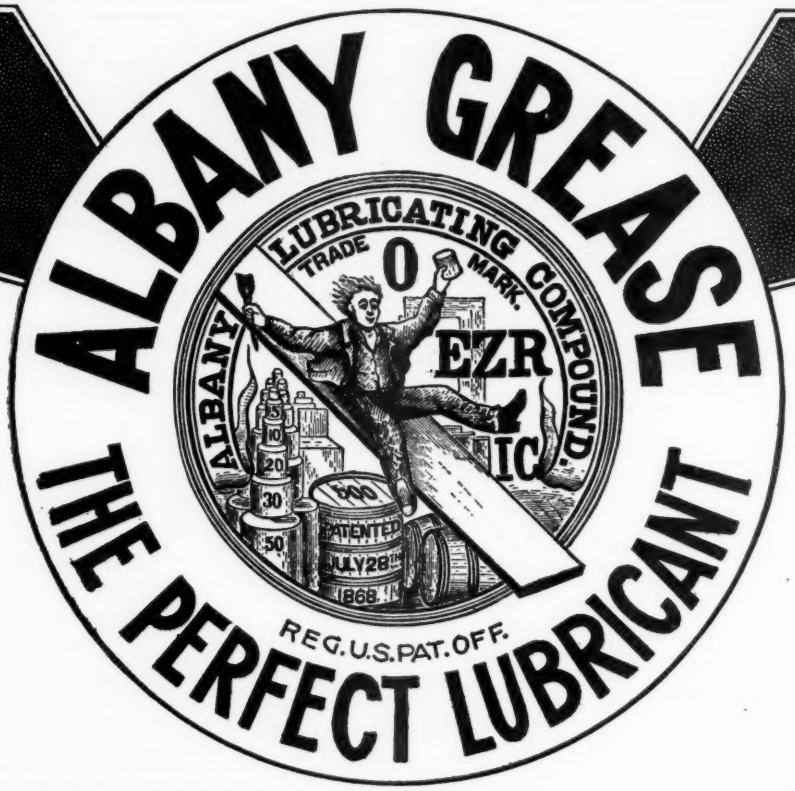

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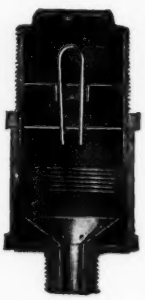

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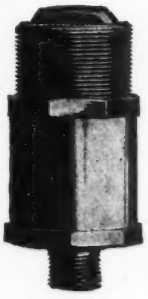

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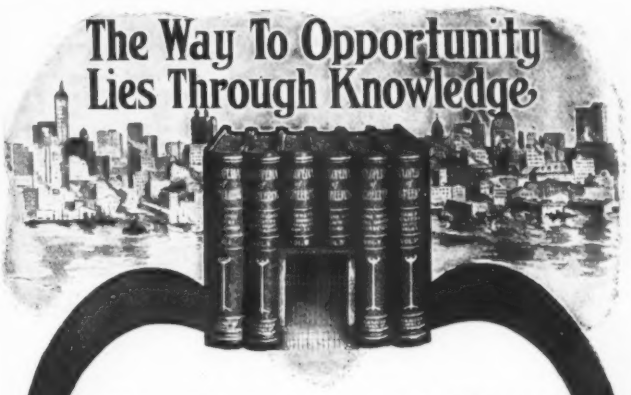

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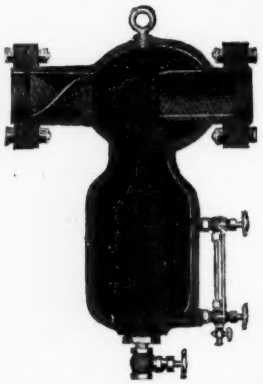

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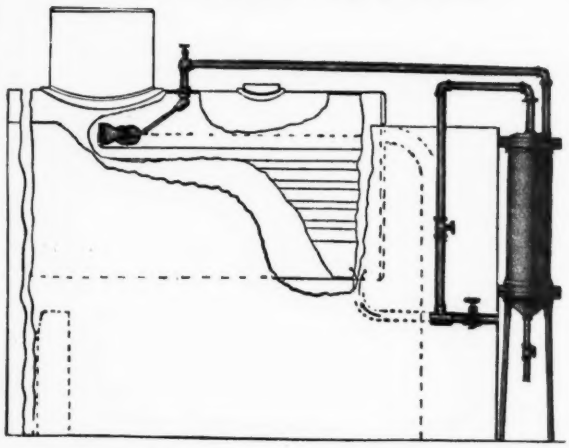

Would it be wiser to swallow it and have a surgical operation to get rid of it, take medicine to dissolve it, or remove it before swallowing?

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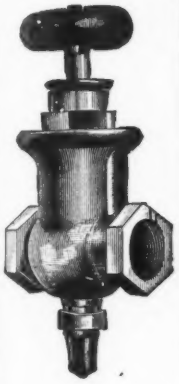

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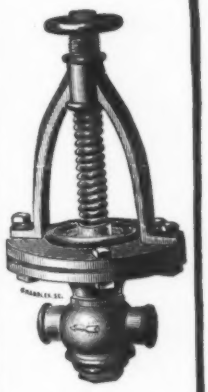

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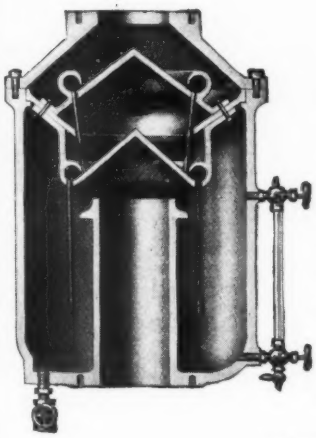

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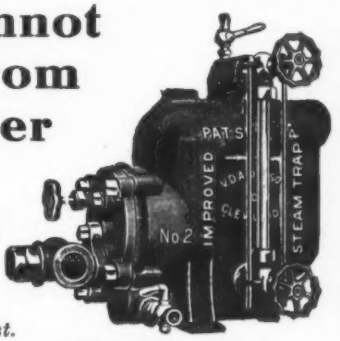

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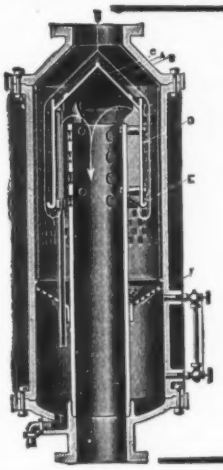

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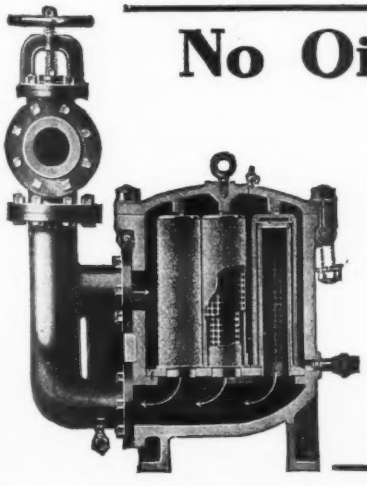

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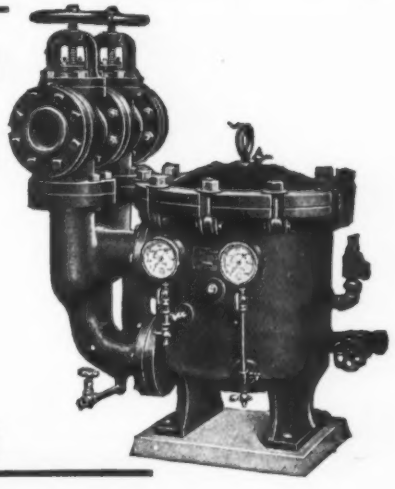

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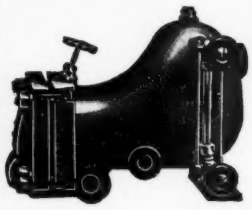

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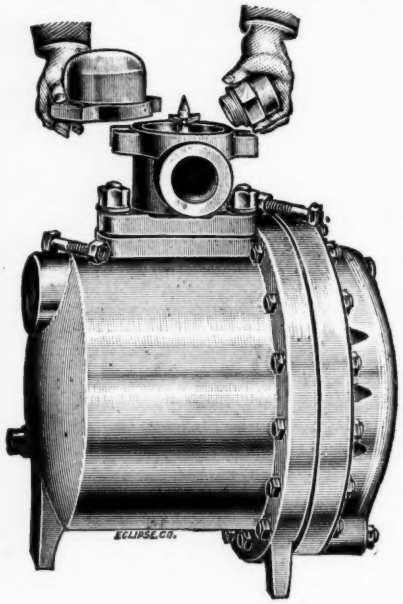

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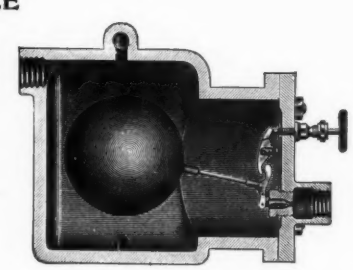

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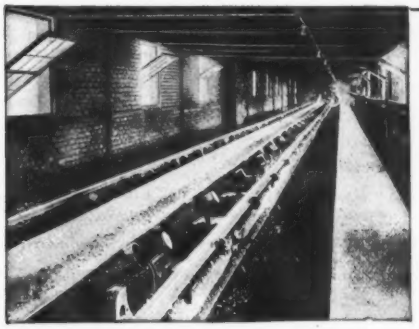

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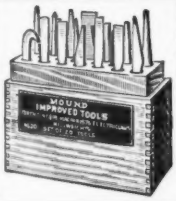

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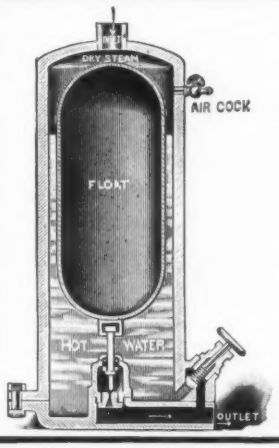
JOHN PLATT & COMPANY
97 Cedar Street, NEW YORK.



The H H Steam Trap

Simple in construction and operation, least liable to get out of repair. Reasonable in price and guaranteed to give satisfaction. Send for Catalog of Pipe Repairs and Steam Specialties.

James McCrea & Co.,
61-65 W. Washington St., CHICAGO

The Law of Gravity

operates the Morehead Non-Return Steam Trap. All working parts are on the outside and can be reached INSTANTLY. The interior is free from VALVES, FLOATS or BUCKETS. We guarantee absolute reliability at all times. Write for Catalog.

Morehead Manufacturing Co., 1051 Grand River Ave., Detroit, Mich.

Robins Laminated Leather Belting

Is made continuous without laps, plies or cement. Guarantees an equal distribution of strength and an uninterrupted line of power transmission.

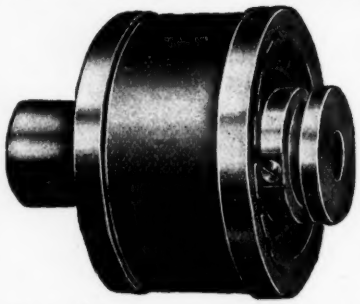
Write for circular No. 2.

ROBINS NEW CONVEYOR COMPANY

New York, 38 Wall Street | Factory, 168 Duane Street | Chicago, Old Colony Bldg.

SAGINAW MANUFACTURING CO.,
SAGINAW, MICH., U S. A.
MANUFACTURERS OF
GILBERT WOOD SPLIT PULLEYS.
New York Branch, 88 Warren Street.
Chicago Branch, 32 South Canal Street.
Sales Agencies in all principal cities.
Cable Address, "Engrave" A. B. C. and Lieber's Code.

AKRON CLUTCHES



Hundreds who tried out other clutches that failed to make good, found just what they wanted when they installed the Akron.

It saves power, expensive shut downs and accidents to employes. Works perfectly at high speeds and starts the load gradually and quietly.

Try the Akron and you won't have to experiment with others.

Write to

The Williams Foundry & Machine Co.,
Successors to the Akron Clutch Co. **Akron, Ohio.**

"DUXBAK" WATERPROOF

DUXBAK BELTING
TRADE MARK

Leather Belting can be soaked in soapy water any length of time without its joints or fibre being affected.

"DUXBAK" STEAMPROOF
can be boiled without being injured in the least. Either sent on approval for your test.

Chas. A. Schieren Company
ESTABLISHED 1868

Tanners, Belt Manufacturers. **69 Cliff St., New York.**

Chicago, 84-86-88 Franklin St. Pittsburg, 240 Third Ave. Boston, 136-138 Lincoln St. Denver, 1622 Wazee St. Philadelphia, 226 North Third St. Brooklyn, N. Y., 13th St. and 3rd Ave. Hamburg, Germany, Auf dem Sande 1. OAK LEATHER TANNERIES, Bristol, Tenn.

Belting Logic—If best Leather Belting costs per sq. ft. $1.00, best Rubber Belting 60c., and best Stitched Canvas Belting 40c., and with Leather showing 25% and Rubber 50% of the strength of **Best Stitched Canvas Belting,** wouldn't it be good policy for you as a practical man to investigate the merits of our product? Send for Book "C."

SAWYER BELTING COMPANY, **CLEVELAND, OHIO.**

THERE'S NO LETTING GO WHEN OXOILOX GETS BUSY.

F. S. WALTON CO., Philadelphia.
Gentlemen:—Kindly send us free of all charge, a sample can of OXoilOX, the perfect belt dressing.

Name..... Name of Firm.....

Address..... Number of Belts.....Average Size.....

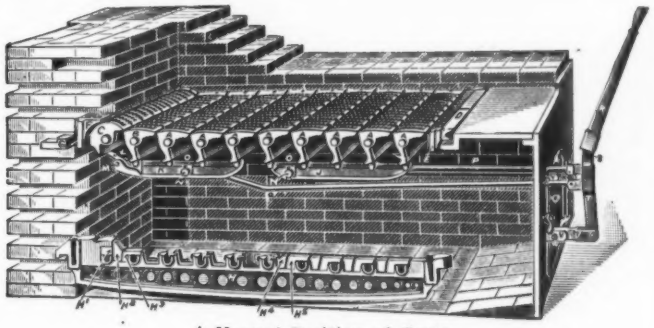
F. S. WALTON CO., Philadelphia, Pa.

Pressers and Refiners of all grades of Neatsfoot Oil.

BUT IT'S THE BELT THAT DOES THE GRIPPING

If a belt has been thoroughly treated with **OXoilOX** you can't make it slip. Run it at full load and full speed—it clings to the pulleys like a chain to the sprocket. **OXoilOX** gives new strength to **old tired belts.** It drives out the "sticky stuff" with which they are usually saturated; makes them clean, pliable and tough. They do **more work** and do it **better.** For **new belts** the best possible start in life is a treatment with **OXoilOX.** It takes the stiffness out; restores the vitality lost in tanning, and helps them get right down to work. Whether old or new, a belt treated with **OXoilOX** adheres so closely to the pulleys that air is absolutely excluded and it's the vacuum thus formed that enables it to **run slack at full load without slipping.** Write for sample.

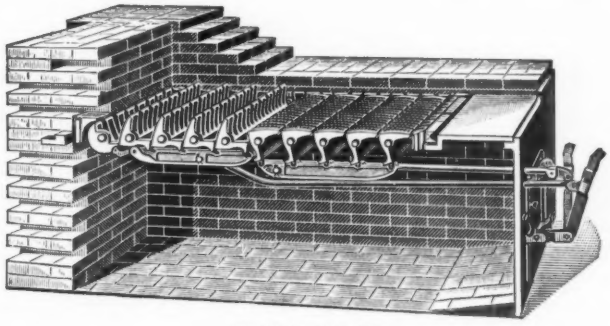
FUEL BILLS TOO HIGH?



Normal Position of Grate

Wouldn't you save a large amount of money if you could burn a cheap grade of fuel and still get high class results?

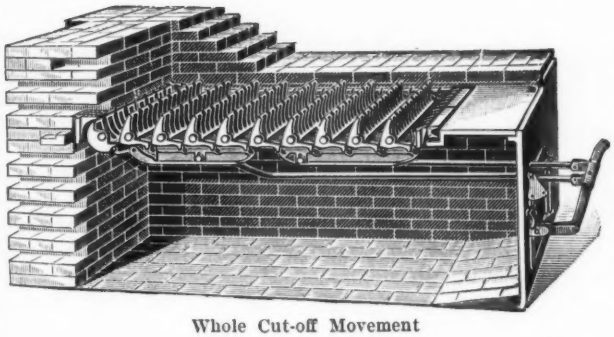
Shaking Movement



Divided Cut-Off

The McClave Grate and Argand Blower

is a system designed to burn such fuel as anthracite, birdseye, rice, bituminous, slack, screenings, duff, etc., and get the same results as other grates burning expensive coal. Complete combustion is secured by the use of the Argand forced blast, and every heat unit is extracted from whatever fuel is used. There's a big saving coming your way if you install the McClave System.



Whole Cut-off Movement

Catalog "D?"

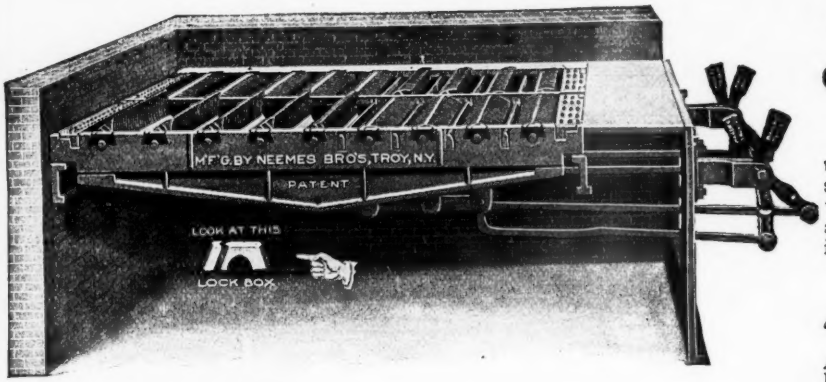
McCLAVE-BROOKS COMPANY,

Scranton, Pa.

BRANCH OFFICES:

Chas. N. Hays, Sales Agent, Fisher Bldg., Chicago; Empire Bldg., Pittsburg.

Argand Steam Blower



NEEMES BROS.

IMPROVED

Clinker Cutting, Shaking and Dumping Grate

unlike all other grates will cut from both sides of the shaker alike. Makes more steam with cheaper fuel and less of it. A slight agitation shakes out the ashes or a full opening dumps everything into the ashpit. *Write.*

NEEMES BROS.,

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C. W. VAN BLARCOM, Room 533, Terminal Bldg., 30 Church St., Agent for New York City and vicinity. BABCOCK & WILCOX, Ltd., Montreal, Canada, Sole makers for Canada. BURKE ENGINEERING CO., 311 Industrial Bldg., Providence, R. I., Sole Agents for the New England States.

PERFECT COMBUSTION PRODUCES NO SMOKE

The Wilkinson Automatic Stoker And Smoke Preventer

procures perfect combustion and a consequent saving of 15% in fuel costs, to say nothing of saving the stokers' wages.

WRITE FOR A CATALOG.

The Wilkinson Manufacturing Co.

Bridgeport, Montgomery County, Pa.

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We are publishing the best books for engineers—we handle all books of all publishers. When you get up against a knotty question, write us and if there is a good book to help you out, we can supply it.

HILL PUBLISHING COMPANY,

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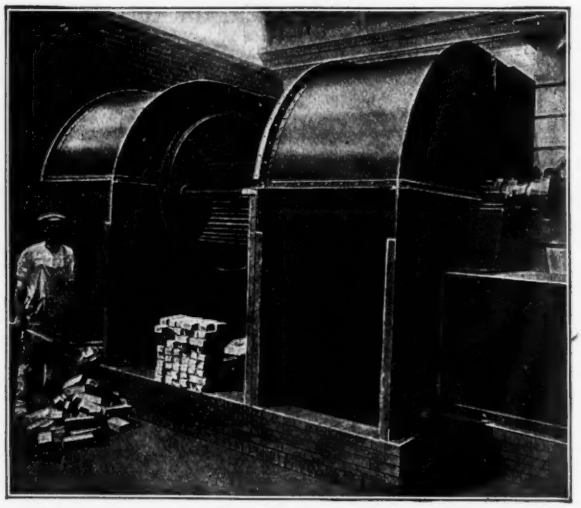
"SIROCCO"

TRADE MARK

BLOWERS

BUILT ON THE TURBINE PRINCIPLE
MOST EFFICIENT FOR ALL DUTIES



VENTILATING INSTALLATION, HOTEL ASTOR, N. Y.
Two 66 inch Sirrocco Fans running on one shaft. Capacity: 50,000 cu. ft. per minute each against ¾ in. W. G. Power saving 25%.

SIROCCO ENGINEERING CO.

138-140 Cedar St., New York. Works: Troy, N. Y,

Chicago Office, 510 Fisher Bldg., T. Lindberg, Mgr.
Pittsburg Office, 1415 Keenan Bldg., F. K. Potter, Mgr.

How do you do your Hoisting

In Yale & Towne Chain Blocks

are embodied the final development in lifting mechanism and the highest integrity in Chain Block construction. They lead in efficiency, speed, durability, reliability and universal adaptation to every hoisting requirement.

Triplex Blocks—½ to 40 tons. Universally recognized as the most durable and efficient Chain Block made. Have balanced train of spur Gearing with large internal gear.

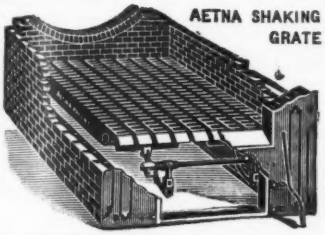
Duplex Blocks—½ to 10 tons. The handiest and safest Screw Blocks made. Have safety guides to prevent slipping of load chains. Bronze Worm wheels and steel worms with hardened and ground thrust bearings running in oil.

Differential Blocks—⅛ to 3 tons. Not as durable or easy lifting as the Triplex or Duplex, but safe and twice as durable as the cheaper kind of the Weston Differential Block, because of superior Chain and Sheaves.

Y. & T. Blocks, duplicate parts and trolleys are carried in stock by Hardware, Machinery and Mill Supply Dealers. Write for catalog.

The Yale & Towne Mfg. Co.,

9 Murray Street, New York.

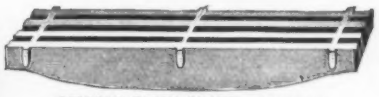

AETNA SHAKING GRATE

SALAMANDER GRATES

Twenty different styles, all made of same grade of iron—the best for the purpose. On the market for 55 years. Tried and true.

SALAMANDER GRATE BAR CO.
126 Liberty Street, New York

ROGERS STATIONARY GRATE


COMMON PLAIN GRATE—CLUSTER

For Forced Draft

there's nothing equal to WING'S TURBINE BLOWER. Does away with tall, expensive stacks and *makes* cheap fuel burn with paying results. Catalog?

L. J. WING MFG. CO., 90 West St., Cor Cedar St., **NEW YORK.**

ANY TYPE OF BOILER

The Murphy Automatic Smokeless Furnace

burns slack and other low cost fuels with unexcelled economy and absolute smokelessness. Strictly automatic in feeding and distribution of coal and the removal of the ash. Send for a complete description.

MURPHY IRON WORKS, 5 Walker St., DETROIT, MICH.

UNITS OF ANY SIZE

DON'T CROWD the brickwork back of your boiler. The Monarch Boiler Arch hangs from above. Easy to set, remove and reset. Money back if not fully satisfied.

Booklet on request.

MAYVILLE SPECIALTY MFG. CO.,
MAYVILLE, WISCONSIN.

MARTIN GRATES

Martin Anti-Friction Rocking Grates will burn slack coal and get the same steam results as ordinary grates do with best lump coal. Catalog tells why.

MARTIN GRATE COMPANY, 281 Dearborn Street, CHICAGO.

GREEN CHAIN GRATE STOKERS
AUTOMATIC SELF-CLEANING SMOKELESS

GREEN ENGINEERING CO.
COMMERCIAL NATIONAL BANK BLDG., CHICAGO
GENERAL FOUNDRY WORK A SPECIALTY

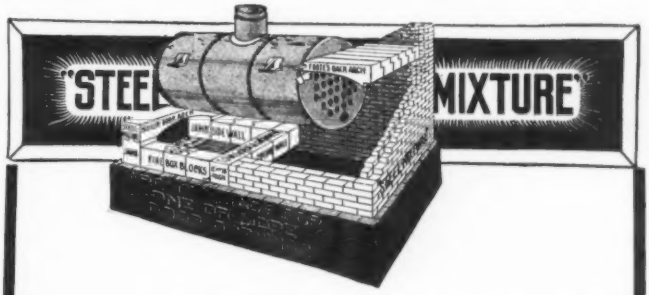
CHIMNEYS

WIEDERHOLDT CONSTRUCTION CO.
American Trust Bldg., Chicago, Ill.

PREVENT SMOKE

Write to us for descriptive matter relating to a device that is mechanically correct, simple in construction, absolutely automatic, and its effectiveness in preventing smoke from Bituminous coal under any and all conditions has been fully demonstrated.

CLIMAX SMOKE PREVENTER COMPANY
205 Equitable Building, Boston, Mass.



A BACK ARCH

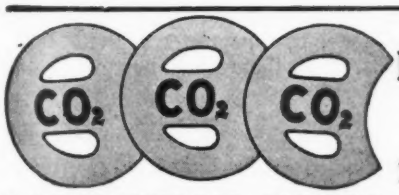
to be a success must allow the boiler to expand freely. Hence it must be supported independently. Foote's Back Arch rests on an iron bar which spans the combustion chamber. The bar is imbedded in the arch blocks and thus protected from the action of the hot gases.

These gases cannot escape because the arch is tight. The shiplapped joints are calked with asbestos and there is an asbestos cushion between the arch and boiler.

The inner surface is flat so that you can use a tool on the upper row of tubes without taking down the arch.

We aim for structural perfection in all our boiler settings. Get our "Dimension Booklet" and let us quote you prices.

McLeod & Henry Co., Troy, N. Y.



C-O-Two Tile
FIRE ARCHES
Any Rise, Any Span,
Air Baffle Mixing System
produces
RESULTS

C-O-TWO FURNACE COMPANY, Syracuse, N. Y.

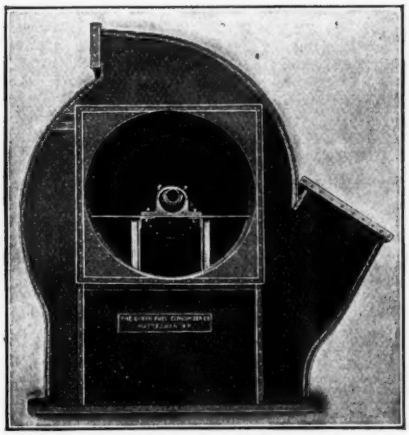
CONTRACTORS.
CENTRAL STATION HEATING.

Write for valuable information.

AMERICAN DISTRICT STEAM CO.,
Lockport, N. Y. Toronto, Can. Chicago.

Over 7500 Hammel Oil Burners
In use under Stationary Boilers, Locomotives, Kilns, Etc.
Write for Circular of Hammel Patent Furnace for Water-Tube Boilers

HAMMEL OIL BURNER CO.
609 North Main Street, LOS ANGELES, CAL.



This Green Ventilating Fan

has two outlets; one "angular-up" discharge and the other "top-horizontal," to save piping and elbows, and fit it to the particular place where it is to be put. A fan "from stock" or from stock drawings might have been cheaper, but would have cost the purchaser more than the difference for extra piping, for which, moreover, there was no room to spare.

It costs nothing to consult us about the layout of air-moving equipment, and our suggestions often save more than the cost of the fans themselves. Our engineering services are free to all prospective buyers or users.

The Green Heater may be used with hot water or steam, or with cold water in the summer for cooling. The advantages of this heater over others are:

The tubes are straight. By taking off two headers you can look through and clean every tube. This is handy where greasy steam or dirty water is used. A defective tube can be at once plugged or removed. The tubes are ordinary one-inch steam pipe, which can be bought anywhere and inserted by anyone who can expand a tube. The circulation is positive from one end to the other through all the tubes. There is no chance for certain short tubes to take all the steam, while longer tubes are inactive, as in older styles. There are no air pockets and no air binding. There is no water hammer, no matter how quickly the steam is turned on.

Our New Fan Catalog "104 P" describes model plants and a full line of Fans, Blowers and Exhausters. It contains useful engineering information. Send for it and our Economizer Catalog, if you do not have a copy.

The Green Fuel Economizer Co.

MATTEAWAN, N. Y.

New York City, Boston, Chicago, Atlanta, San Francisco, Los Angeles, Seattle, Salt Lake City, Montreal.

Engineers; Builders of Green's Fuel Economizers, Fans, Blowers and Exhausters, Steam Air Heater Coils, Waste Heat Air Heaters, Mechanical Draft, Heating and Ventilating and Drying Apparatus, Draft Dampers and Engines. 74

Every Coil a Spring

Reilly Multicoil

FEED WATER HEATER

IN AN ADVERTISEMENT we cannot describe all the advantages of this heater. We therefore ask you to write for our Catalog.

The GRISCOM SPENCER CO.

90 WEST ST., NEW YORK.

The Reilly Oil and Grease Exractor. | Agents: Russel Engines—Tudor Boilers Mfgrs: Tanks—Special Machinery of all kinds

TACHOMETERS

Our No. 4 Hand Tachometers will register speeds in either direction from 30 to 4,000 revolutions per minute. Only one spindle required for all four speed ranges. The change from one speed range to another can be effected while the Tachometer is in use, by pushing the thumb slide on the face up or down. *Send for Catalogue.*

Schuchardt & Schutte,

136 Liberty Street, - New York.

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Sign and mail the coupon below. Send no money! Take no risk!

One hundred and twelve of the world's master business men have written ten books—2,079 pages—1,497 vital business secrets, ideas, methods. In them is the best of all that they know about

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The System Co., 151-153 Wabash Ave., Chicago

If there are, in your books, any new ways to increase my business or my salary, I should like to know them. So send on your 16-page free descriptive booklet. I'll read it. 183-15

Name_____

Address_____

Business_____

Position_____

"A HEATER—AND A RE-HEATER"

ERIE CITY OPEN TYPE FEED WATER HEATERS

contain our own especially designed inlet head and trays.

They have a deflecting plate which prevents any water, coming in contact with the sides of the heater, from flowing out through the overflow.

These are but two of the many exclusive features, *all of which are patented*, that our heaters contain. They are all fully explained in a splendidly illustrated catalogue.

Send for one, its FREE.

ERIE CITY IRON WORKS,
ERIE, PENNSYLVANIA.

Sane Economy

You *must choose* between wasting your exhaust steam, while buying coal to heat your feed water *and* using the exhaust to save that amount of coal

The Otis Tubular Feed Water Heater and Oil Separator

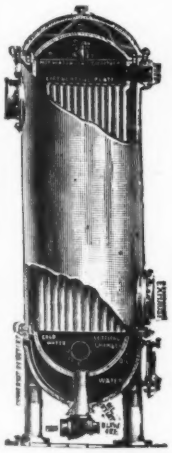
is guaranteed

The most Easily cleaned.
To heat feed water to 210° without back pressure.
To remove scale forming impurities and extract the oil.
To save 15 to 25% over an injector.

Let us tell you all about it.

The Stewart Heater Company,
45 E. Delavan Avenue, Buffalo, N.Y., U.S.A.

Riblet Feed-Water Heaters



All Tubes Straight
and made of the best heavy drawn seamless brass tested to 500 pounds.

Powerful Shells
of cast iron or boiler plate, depending on the size of the heater.

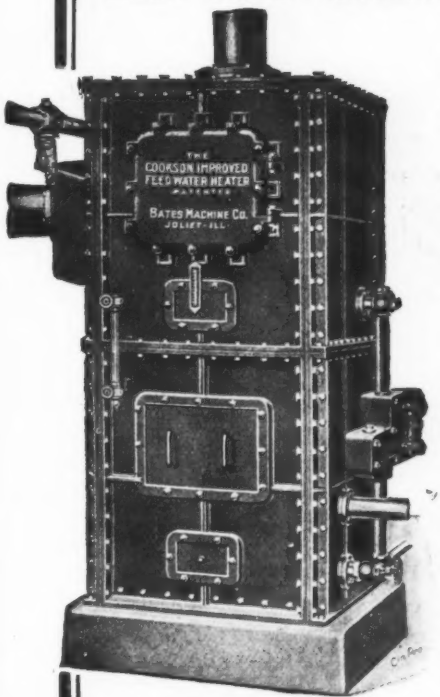
Simple Construction
makes them easy to get at if repairs or cleaning should become necessary.

The steam area is so great as to eliminate back pressure.

Ask for other details.

Erie Mfg. & Supply Company,
1203 Peach Street, ERIE, PA.

COOKSON STANDARD ALL CAST IRON FEED WATER HEATERS



Highest Temperature and Positive Purification of feed water.

Perfect Separation and Removal of Oil.

Automatic Control of Cold Water Supply.

Dry Steam for Heating Purposes.

Finest Grade Special Fittings.

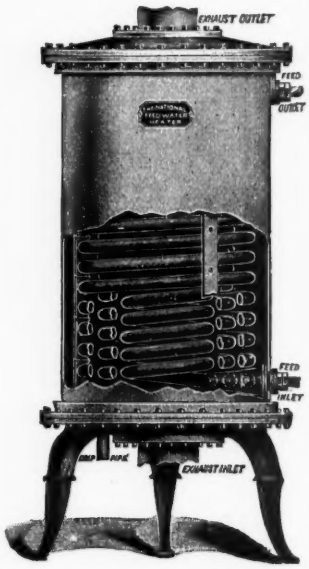
Lime Catching Pans. Especially Adapted for use with Heating Systems, Dryers, Etc.

Write for our new Heater Catalog.

BATES MACHINE COMPANY
Sole Manufacturers **Joliet, Illinois**

Exhaust Steam Costs Money Make It Work!

Get a National Feed Water Heater—it consists of seamless drawn brass or copper tubing incased in a cast iron shell. It differs from other heaters in that it costs nothing but exhaust steam to operate it. Over 1,200,000 H. P. in use.

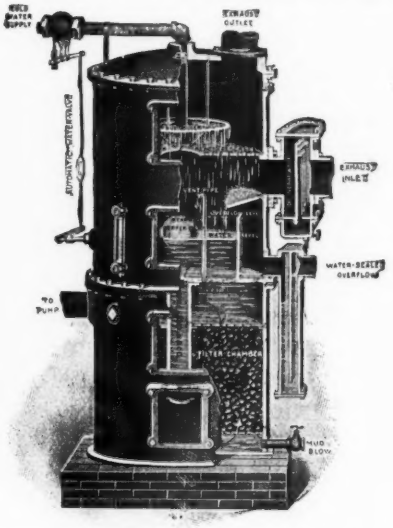


Catalog!

The National Pipe Bending Co.,
175 Lloyd Street,
New Haven, Connecticut.

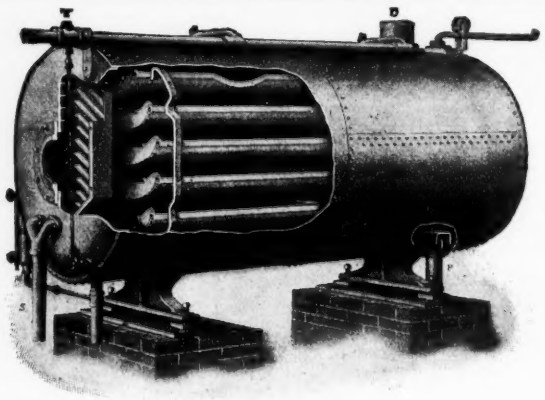
120 Liberty Street, New York. 54 High Street, Boston, Mass.

THE WHITLOCK OPEN EXHAUST Feed Water Heater and Purifier



Perfect Oil Separation. Easily Cleaned. The Highest Possible Temperature in the Feed Water.

Manufactured by
THE WHITLOCK COIL PIPE COMPANY,
HARTFORD, CONN.
New York Office—Singer Building, 85 Liberty Street.



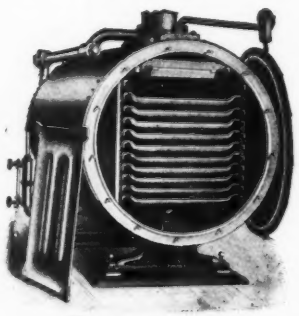
Standard Steel Construction Heater

HOT FEED-WATER NOT "HOT AIR"

is what engineers and steam users desire when looking for a feed-water heater. We GUARANTEE the former to the highest possible degree where our heaters are installed. The latter is not in our line. *But just a fact or two.*

HOPPES FEED-WATER HEATERS and PURIFIERS

contain a much greater amount of heating and lime-catching surface than any other heater of equal rated capacity on the market. They bring the water into direct contact with the steam in thin films on this large heating surface and as a result give the highest possible temperature obtainable from exhaust steam, removing all scale-making impurities that will crystalize at this temperature. If you want to know more WRITE FOR OUR CATALOG.



Heater—Class "R"
Cast Iron Construction

THE HOPPES MANUFACTURING CO.
19 LARCH ST., SPRINGFIELD, OHIO.

Harrisburg Feed Water Heaters

Of Pure Seamless Copper Coils. Guaranteed to be the most effective, most durable and cheapest heater manufactured.

Please write for Descriptive Catalog.

COPPER, IRON AND BRASS PIPE COILS AND BENDS OF ANY DESIRED SHAPE

Manufactured by

The Harrisburg Pipe & Pipe Bending Co.

950 HERR STREET, HARRISBURG, PA.

Manufacturers of HIGH GRADE

Wrought Iron

PIPE

BLACK AND GALVANIZED

NO SCALE can form if the feed-water be heated to 300° before entering the boiler. Yes!— we use the live steam to do it. but we save over all other methods—and we can prove it. Catalog ?

The Eclipse Feed Water Heater & Purifier Co.,
Oshkosh, Wisconsin.

BURNHAM

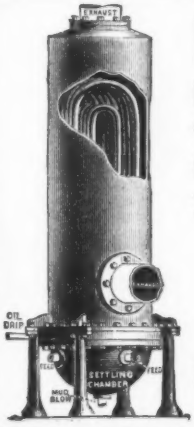
STEAM PUMPS

ALSO

POWER PUMPS—AIR COMPRESSORS—CONDENSERS

UNION STEAM PUMP CO.,

BATTLE CREEK, MICHIGAN.



Puff—Puff—Puff

Haven't You Lost Money Enough

That exhaust steam will save fuel, equipment and repairs if you turn it into a

Patterson-Berryman Feed Water Heater and Purifier.

Write for particulars of our 60-day trial offer

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COLD WEATHER FANS

The exhaust fan is as necessary in winter as in summer, for with closed windows natural ventilation is restricted. The propeller fan can be moter-driven or belted.

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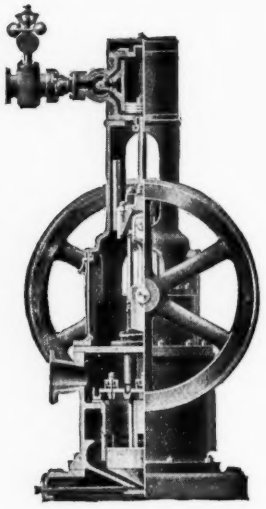


POWER CONSUMPTION of the WHEELER-EDWARDS PUMP

Before discussing the matter, let it be understood that we are willing to guarantee a lower steam consumption while producing a given vacuum under a given set of conditions with the Wheeler-Edwards Air Pump than can be obtained with any other type of wet vacuum pump, or with any combination of dry vacuum pumps and hot well pumps.

Theoretically considered, how much power does it take to remove the water from a surface condenser and to compress and remove the air? If you will figure it up, you will find that it is hardly noticeable as compared with the output of the main engine.

What then causes the high power consumption of many air pumps? The answer is, mechanical friction of the plunger and other moving parts and fluid friction of water churned back and forth. In dry-vacuum pumps most of the power is wasted in the cylinder, where the piston usually has snug-fitting rings and rides in a horizontal position on one side of the cylinder. In most wet vacuum pumps you have the same evils, plus those due to churning a large volume of water repeatedly back and forth; the effort of the latter is especially noticeable at high speeds.

In the Wheeler-Edwards Pump the vertical plunger or piston works in a vertical barrel with water packing, and the result is almost entire absence of friction. No water is churned back and forth, as all of the charge admitted at each stroke is delivered at the end of that stroke, no matter how slow or fast the pump is run. Some people have thought that because an almost perfect vacuum is created in the pump chamber during the descent of the piston, after which the pump chamber is then suddenly in communication with the condenser, some power would be wasted in this way. However, if you consider that the difference in pressure on the two sides of the pump piston is only that due to the AIR in the condenser and that the pressure of the latter is probably less than ½-in. mercury, you will see that the power lost in this way is almost too small to calculate. In fact, the Wheeler-Edwards pump operating under normal conditions shows a card of exceptionally small area. (See POWER, p. 560.)

The Wheeler-Edwards Air Pump not only uses very little power, but what it does use it makes itself in an economical manner, that is, it is fitted with a steam cylinder having a valve gear admitting of setting the cut-off at the most economical point.

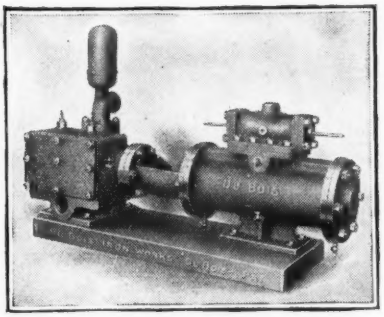
We shall be pleased to give you full particulars on this subject, if you will communicate with us. Ask for Book P-103.

We build Wheeler Surface, Wet and Barometric Condensers, Wheeler-Volz Combined Surface Condensers and Feed Water Heaters, Wheeler Feed Water Heaters, Barnard-Wheeler Cooling Towers, Wheeler-Edwards Air Pumps, Wheeler Centrifugal Pumps, Wheeler Rotative Dry Vacuum Pumps and Wheeler Multiple Effect and Evaporating Machinery. 31

WHEELER
CONDENSER & ENGINEERING CO.
CARTERET, NEW JERSEY.

DuBois Pumps

ONLY ONE VALVE



and in all other respects

Unequaled For Simplicity

There are no outside valve gears to bend or bind.
All working parts are completely covered.

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Established 1877
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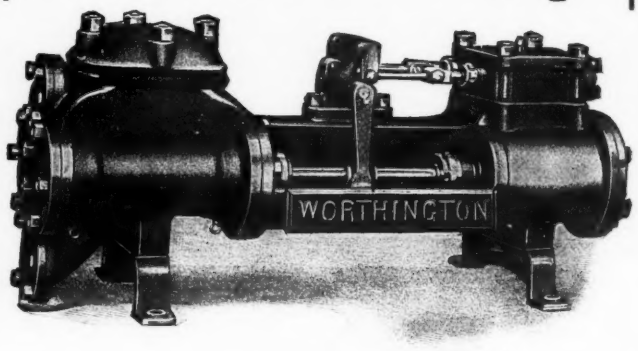
Boiler Feed Pumps



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BEST WORKMANSHIP
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Works, Harrison, N. J.

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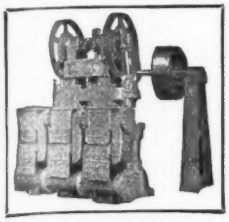
Ask for Catalogue "G" and treatise "Feeding Boilers."



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For operation by any power. **Pumps**

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Steam Pumps

of the Highest Class.

Repairing of Pumps a specialty.

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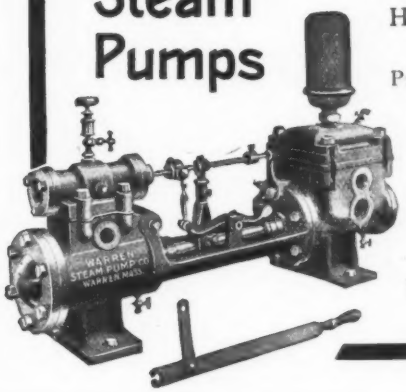

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Dean Bros. Steam Pump Works

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Simple and Duplex Boiler Feeders. Jet and Surface Condensers. Power and Electric Pumps.

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Alberger Pump Co.

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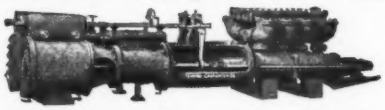

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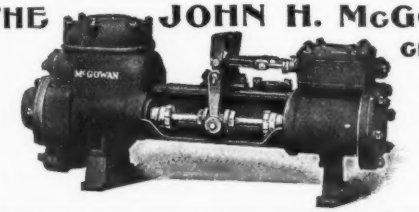
Cincinnati, Ohio



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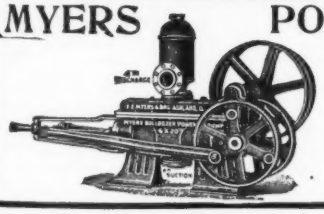

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The **Baragwanath Syphon Condenser** will secure and maintain a vacuum ranging from 24 to 26 inches on engine service and as high as 28 on vacuum kettles, evaporators and stills.

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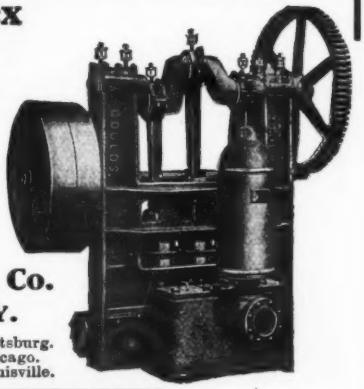

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COPES

Automatic Boiler Feeding Is The Order Of The Day!

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But the movement to adopt automatic regulators is still deeper. No fireman is as trustworthy as a correctly designed regulator, properly installed. Human beings are fallible, while mechanical forces under known conditions are absolutely certain. The history of mechanical improvements shows a steady substitution of automatic for manually operated devices.

An automatic device, of course, must meet certain requirements. It must have no delicate and vulnerable parts, such as springs, pilot valves, diagphrams, floats, etc.; it must be well made and simple. But in any case it makes no errors of judgment, as carrying a high water level to be on the safe side, or turning a valve the wrong way in a moment of abstraction. It does what no man can do, it feeds the water at exactly the right rate to each of any number of boilers, to utilize the fuel saving capacity of feed water heaters and economizers, to the fullest extent.

Now is a good time for the progressive engineer to look into this matter of boiler feed regulators. Send for our new Treatise "P."

You will find that the Copes is the only one that is direct-acting, that has no floats, pilot valves, springs, or diaphragms; the only one with all its working parts always in reach and in plain sight; the only one that is put in on 60 days' trial and guaranteed for 5 years.

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29

The Fisher Steam Pump Governor

Simple in construction, absolutely automatic in operation and easily regulated.

It pays in the end to secure THE BEST. That Fisher Governors *are* best, is shown by the fact that *more of them are in use today than all other makes combined.*

Shall we send *you* one on trial?

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If A Belt Breaks

the Automatic Safety Stop of the **Gardner Standard Governor** closes the valve instantly by removing the support of the speed lever fulcrum.

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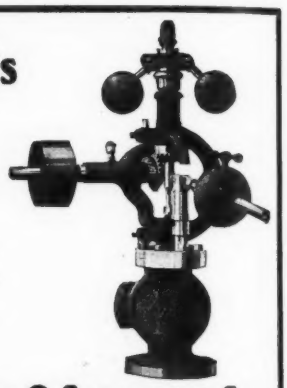
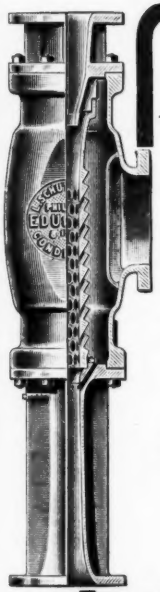


The Gardner Governor & Separator Co.
Quincy, Illinois.

Koerting CONDENSER



The Koerting Condenser is the simplest, efficient and most durable on the market, can be applied to engines and turbines of any size, and works with absolute certainty, producing the same high vacuum under all conditions of load variation.

This condenser, unlike many others, requires no air pumps whatever. The only movable part in a Koerting Condenser plant, is a centrifugal pump to produce a head of water of 8.6 lbs. Space required is exceedingly small. Vacuum 93%. For further particulars,

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FRISCO: O. C. Goerlz & Co.

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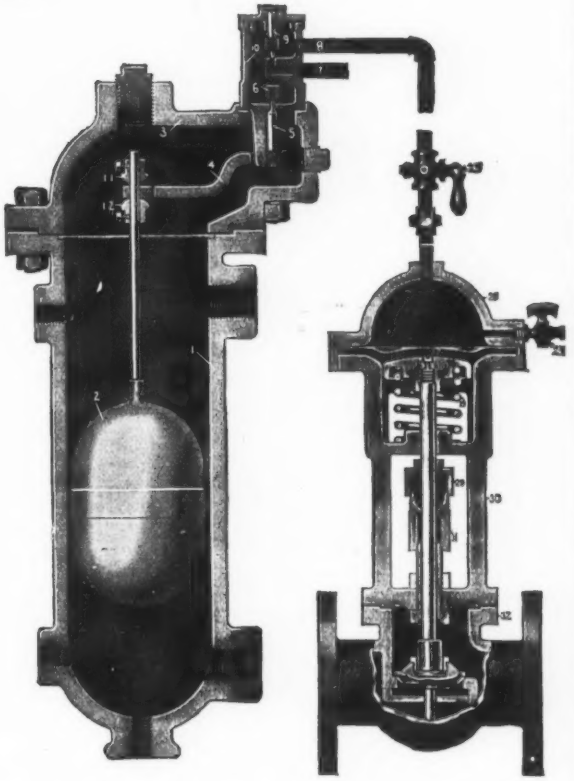
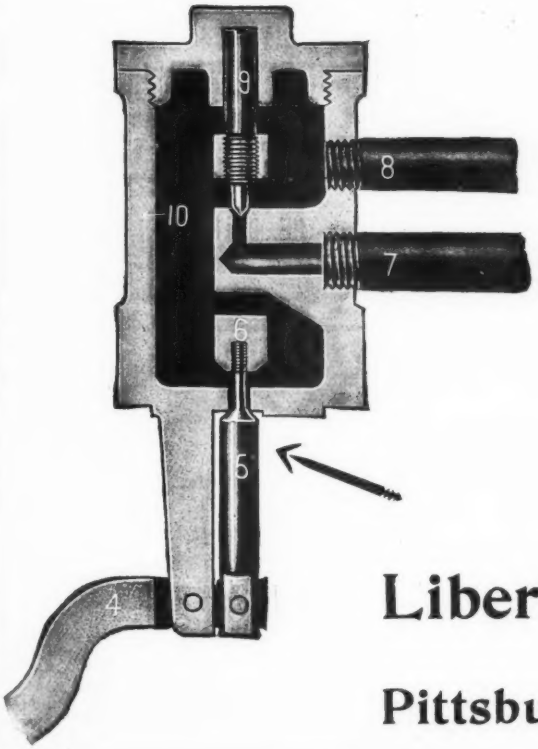
Close Regulation

In these three fundamental features, the **Liberty Feed Water Regulator** is beyond comparison. But its superiority doesn't stop there. For fine material and expert workmanship, it is way ahead of all others.

The operation of the "Liberty" is positive. And note this—no slide valves, gravity-operated valves, thermostats or leakage valves are used in its construction. *The float* controls the pilot, admission and exhaust valve.





Admission Closed
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Note the illustration and *study it.* The pilot valve can be adjusted under *full steam pressure.* The admission valve stem and exhaust valve stem operate simultaneously.

Write for particulars. We manufacture the famous **Liberty Devices**—devices *you need* around your plant.

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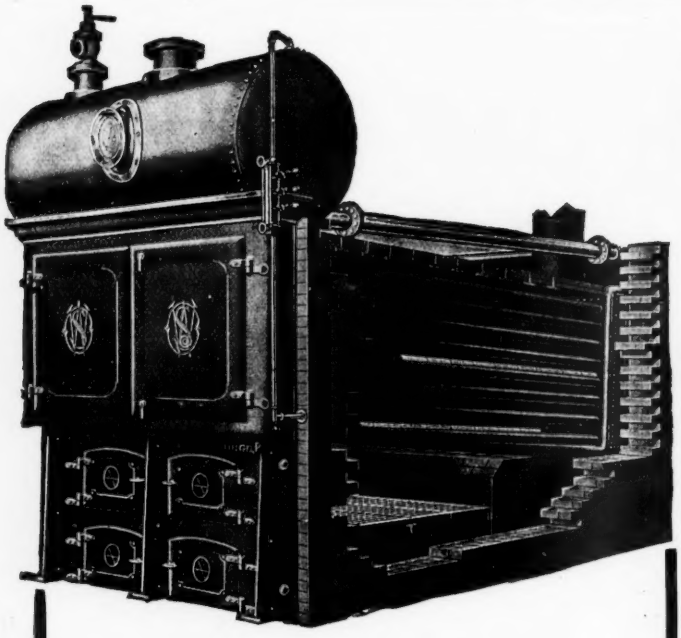

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in any power plant is the boiler system. If you appreciate that fact you will do well to investigate

Our Safety Water Tube Boiler

Built for any pressure from 125 to 300 pounds. Meets all the high duty requirements of modern triple and quadruple expansion and compound engines. Perfect circulation, remarkable economy, and freedom to expand and contract, are features.

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That's the simple and never failing principle upon which the Davis Pressure Regulator operates. It has no springs, toggles, diaphragm or auxiliary valves in its make up. It accomplishes its purpose—the automatic reduction of pressure—by the most direct means and enables you to use steam at any pressure less than that of the boiler for auxiliary apparatus, heating, drying, cooking or any other purpose. When once set for a certain delivery, it automatically maintains that pressure constant regardless of boiler fluctuations.

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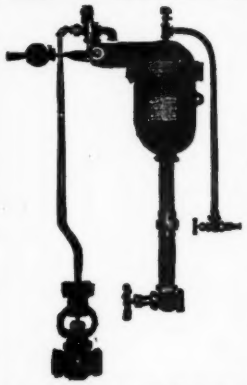

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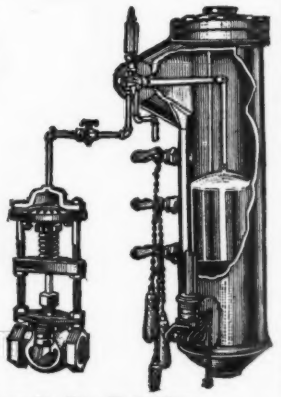
Safety Automatic
Feed Water REGULATOR



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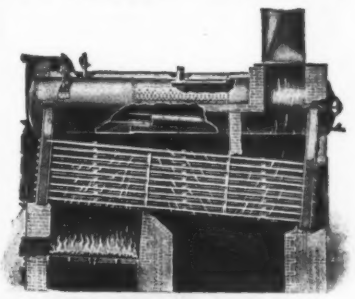

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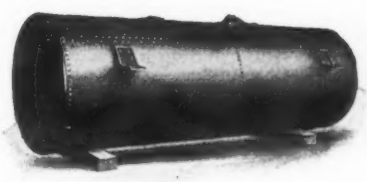

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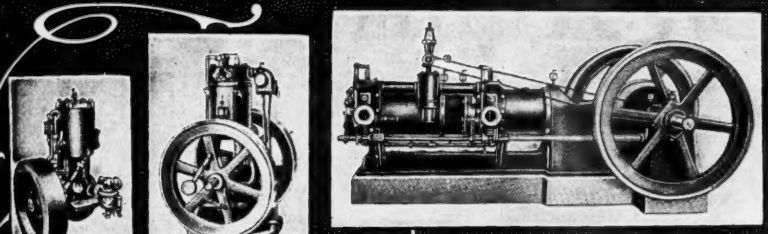

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GAS, GASOLENE, KEROSENE, ALCOHOL
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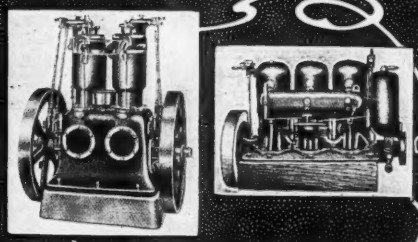
ALL PURPOSES TEN TYPES ALL SIZES



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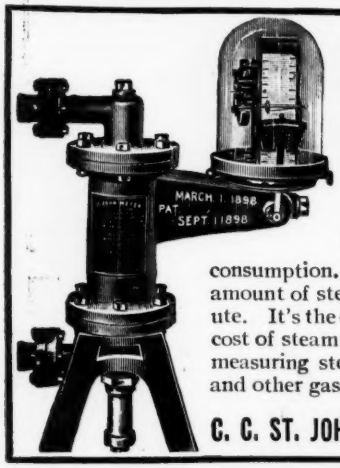

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The St. John Indicating and Recording Steam Meter gives you a daily record of steam consumption. It also has a dial that tells amount of steam being delivered any minute. It's the only way to keep track of the cost of steam. It is not only adapted for measuring steam, but measures water, air and other gases as well. **Want the details?**

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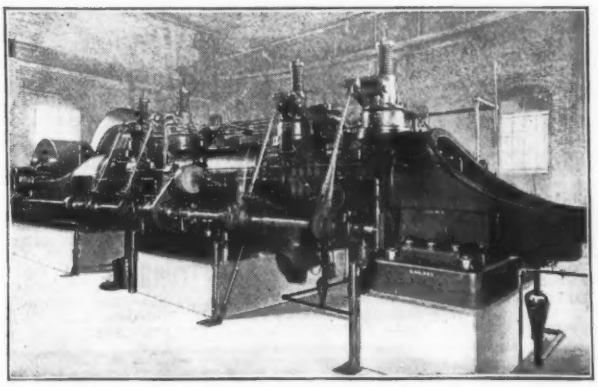

Westinghouse Double-Acting Gas Engine Generating Unit.

The Westinghouse Machine Company

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is a consideration of vital importance with Central Stations in buying wattmeters—not first cost, but ultimate cost—the cost of maintenance. A large number of Central Stations are saving money every year by buying

Type K Meters

They have a record for service, accuracy and reliability back of them. Type K meters have been satisfying customers for years because of these superior qualities. 746

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FOOS

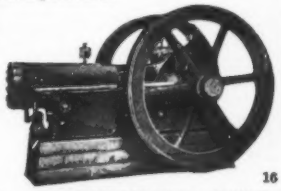
GAS AND GASOLINE ENGINES

HAVE WIPE SPARK IGNITER, positive acting and self cleaning.

VERTICAL VALVES, mechanically operated, admit fuel charge into cylinder at atmospheric pressure.

CENTERLINE COUNTERBALANCING increases the efficiency and decreases the fuel bills.

ACCESSIBLE DESIGN makes it a simple matter to keep Foos Engines properly adjusted.



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16

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The overload capacity of **C-W** electric generators makes them popular with the engineer. In emergencies they do better than is expected of them. Bulletin 80M describes our D. C. machines.

CROCKER-WHEELER COMPANY
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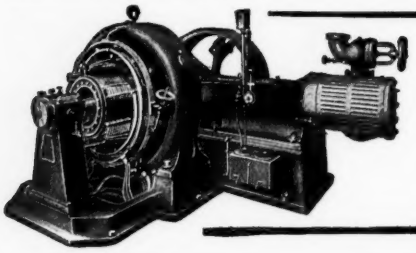

SKINNER ENGINES

THE STEAM TIGHT VALVE KIND

SEND FOR COPY OF "*COAL PILE DIVIDENDS*"—MAILED FREE

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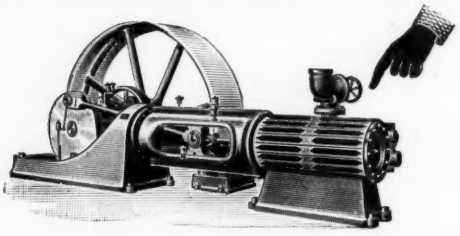

ONLY ONCE A YEAR

and then only for one hour need Fitchburg Engines be worked over to make them steam tight. For 25 *years* they will keep steam tight with only 25 hours work. Catalog 61?

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for light, heat and power purposes from natural gas and Portable Stills for vegetable waste matter. Economy and rapidity of construction a specialty. Unquestionable references.

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See Harper's Weekly, Oct. 3, 1908. Patent Serial No. 450,294—Aug. 25

Make Your Engine Talk

and it will keep you posted on its needs.

THE TRILL TRIUMPH INDICATOR

is a device which makes the engine talk. There are two other kinds of indicators, one costs $50 too much, the other makes the engine tell lies. Better buy the Trill. It is shipped subject to approval.

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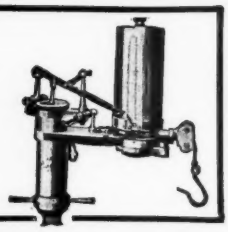

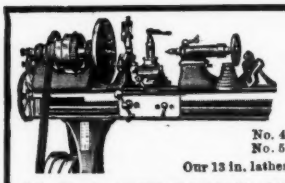

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"YOUNGER TRIUMPH"

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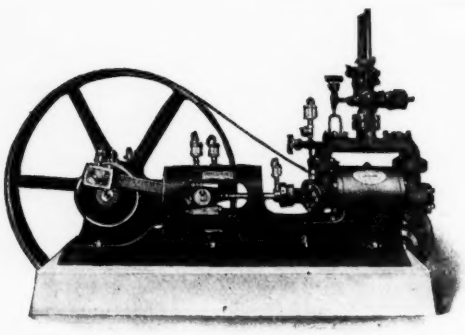

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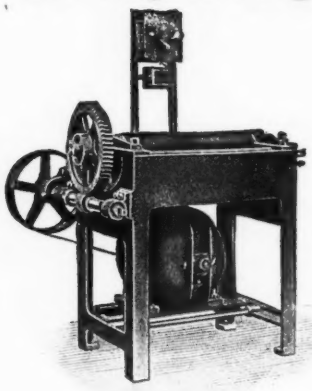

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of all types are so carefully built, to meet every modern requirement, that highest possible steam economy always follows their installation. They are controlled by the Wisconsin Releasing Gear, the only real valve gear improvement in 20 years. Read over that article in Power for Oct. 27th.

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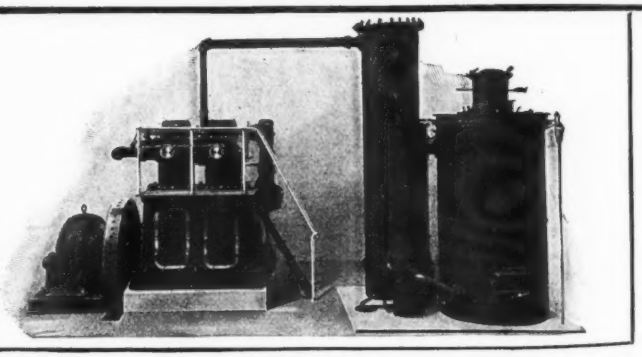

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SATISFACTORY AND CONTINUOUS SERVICE
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FEW IN NUMBER
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ENGINES OF VERTICAL AND HORIZONTAL TYPES
75,000 HORSE POWER IN SUCCESSFUL OPERATION
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BUILDERS OF HIGH CLASS
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HIGH IN DUTY, SUPERIOR IN REGULATION
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ABSOLUTELY RELIABLE
CORRESPOND WITH US BEFORE PURCHASING
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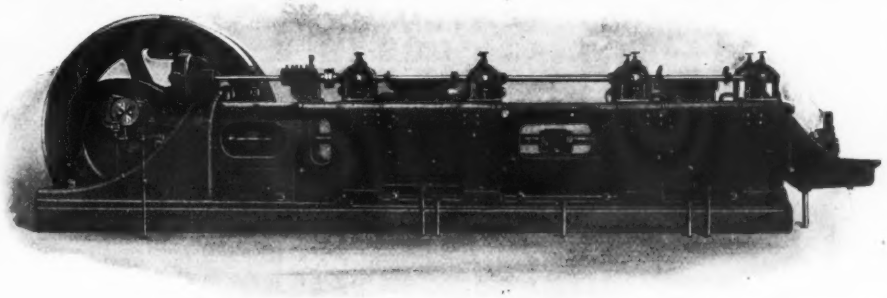

Riverside Heavy Duty Double Acting Tandem Gas Engine (Class "F").

Not Made From Wash Drawings

The cuts we show are made from engines we have built and delivered to purchaser, and they are giving steam engine satisfaction at a fraction of the cost, some of them in twenty-four hours' service. The RIVERSIDE heavy duty gas engine, is wholly unlike any other and is built in twelve types and seventy-two sizes from 10 to 2500 H. P.

We also Build a Complete Line of Gas Driven Air and Gas Compressors

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Take a little of the Polish on a soft cloth, rub the Metal hard and then wipe it off with a dry cloth

THE ONLY METAL POLISH

that works quick and easy and keeps its lustre. Holds old trade and makes new. IT DOES NOT DETERIORATE. Established 16 years.

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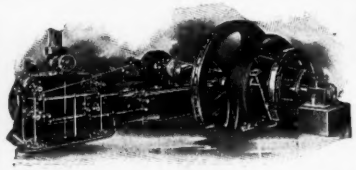

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Reliable and Economical.

Designed and Built by
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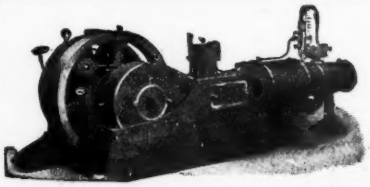

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The plain, simple brute force which can be reckoned in horse power or the convenient, versatile (we may say "intelligent") power afforded by Northern Apparatus? ¶You need better power than ordinary methods afford. Northern Machines and methods are best. They assure maximum convenience in the arrangement of production machines, maximum economy in the application of labor and maximum profits. ¶An intelligent discussion of the use of Northern Machines in your work necessitates that we know what your work requires. Tell us and ask us incidentally for booklet 839.

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A Superior NON-POISONOUS Jointing Material of Metallic Composition; for FLANGE and SCREW-JOINTS of every description on Steam, Water, Air, Gas, Ammonia, etc. UNITES TWO METALS AS ONE. Cheaper, more efficient than red lead—no waste in its use.

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Manufacturers of the Erwood Straightway Swing Gate Valve. Many valuable features in one valve. Write for Catalog.

24 Hours a Day

and 365 days in the year. That's the kind of service one gets from a

SHEPHERD ENGINE

and that's why the name *Shepherd* has come to stand for absolute *reliability* and operating *economy*. Our illustrated catalog free on request.

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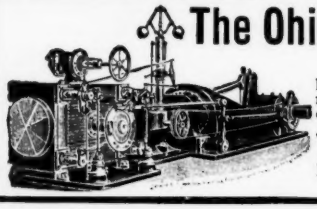
The Ohio Corliss Engine



For hard work, heavy loads and high-speed, the Ohio shows close regulation and runs smoothly and quietly.

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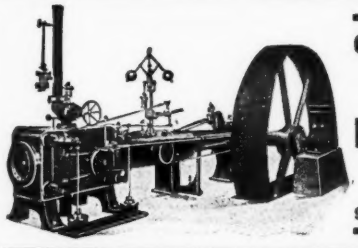

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Refrigerating and Ice Making Machinery

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Its consistent record of success and popularity is not accidental but the result of constant effort devoted exclusively to engine building for the past fifty years. Write for Bulletin No. 15.

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Pacific Coast Representatives: Armstrong-Kreiling Machinery Company, San Francisco, Cal.

We Manufacture
AN ECONOMICAL AND DURABLE ROTARY ENGINE
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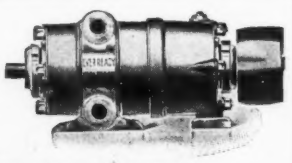

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ALBERGER TANDEM GAS ENGINES
ACCURATE REGULATION FOR ELECTRIC SERVICE-AUTOMATIC CUT-OFF AND RITES FLY-WHEEL GOVERNOR
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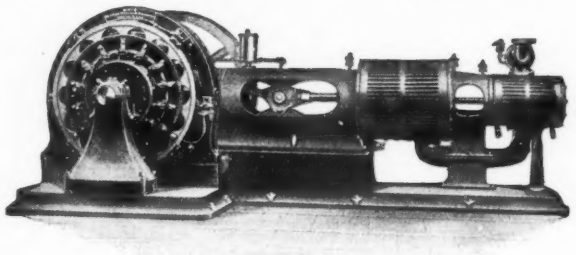
We Publish A Book Filled With Names Of Users



of the Ridgway Dynamos and Engines. A glance through its pages will convince you that the best engineers in the country have installed the Ridgway.

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Ridgway Dynamo & Engine Co., Ridgway, Pa.

Fine Results From Cooper Corliss Engines

are being secured in all classes of service. We are Engine Builders and also do complete Steam Plant Engineering.

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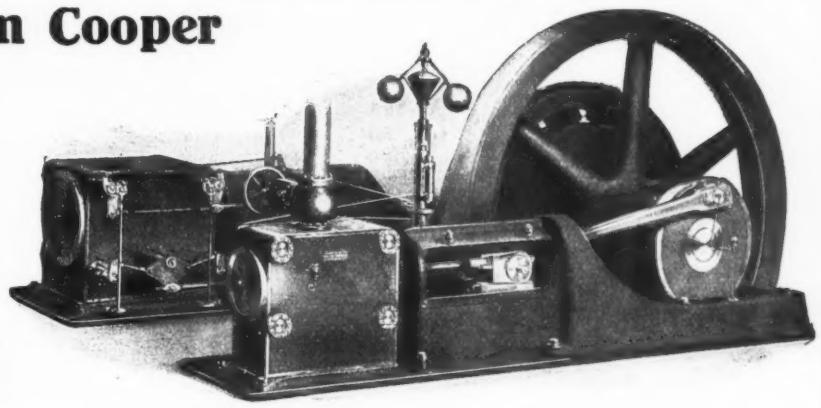
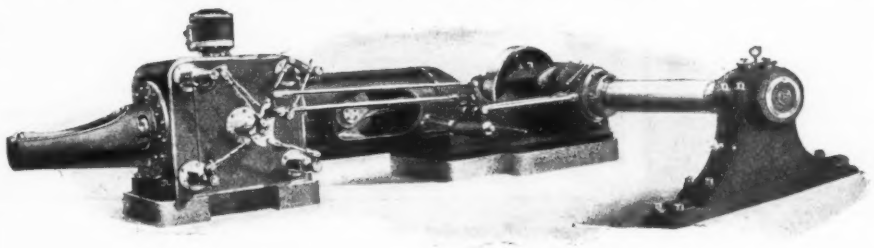


The Perfect High Speed Corliss Engine



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A Different And Better Bearing Metal

Stays where it's put and is frictionless. It has a new efficiency for all factory and machinery bearings and its lasting power is remarkable. Sells for 25c a pound at all dealers, or direct from us in 50-lb. trial boxes. Try it.

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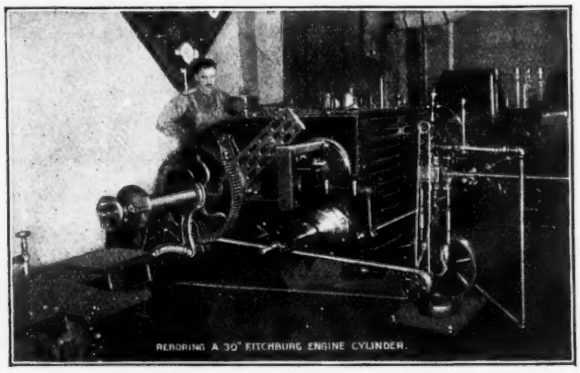

All Kinds Of Repairs To All Kinds Of Engines

That's our business, going all over the country fixing engine troubles of all sorts and reboring cylinders (up to 120") in position.

It doesn't pay to waste time dismantling your engine, breaking connections, and waiting ages for the repaired part to be returned. We do the work *in place*. Write!

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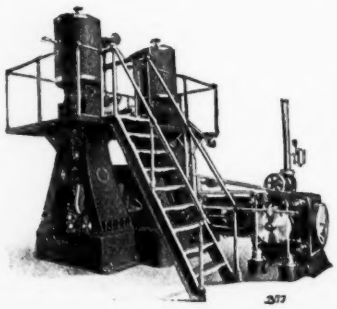

We manufacture all machinery and parts needed to equip a complete Ice or Refrigerating plant—

Machines, Condensers, Tanks, Cans, Coolers, Piping, Boilers and Ammonia Fittings

of all kinds. We employ over 1,250 men in the manufacture of Ice and Refrigerating Machinery exclusively. Catalogue sent upon request.

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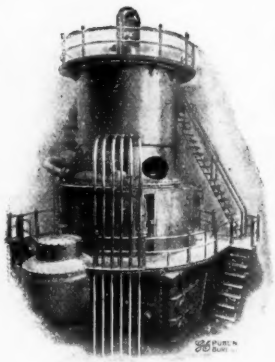
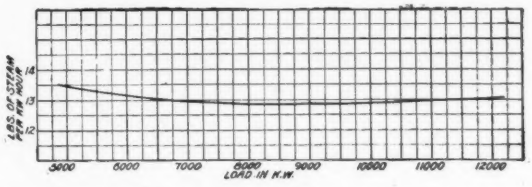


Curtis Steam Turbines show a nearly constant efficiency over a wide range of load.

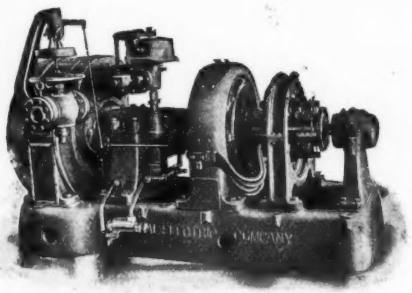
The following load cure is based on tests made in the Fisk Street power house of the Commonwealth Edison Company, Chicago, Ill., and shows a maximum variation in economy of 5 per cent. from 5000 to 12000 kilowatts.



"Load Water Rate Curve" for a 9000 kw., 5-Stage, 750 R. P. M. Curtis Turbo-Generator Unit, Operating at 200 lbs. Gauge Pressure, with 125 Degrees Superheat and 29 Inches Vacuum.

Curtis Steam Turbines are made in standard sizes from 15 kw. to 14000 kw.

The Curtis Low Pressure Turbine is available in standard sizes from 300 kilowatts up, utilizes exhaust steam from 1 pound gauge pressure to 29 inches of vacuum, adds 30 to 100% to the output of steam engine plants and does not require a pound of additional fuel. Some installations have been in successful operation for years.



1696

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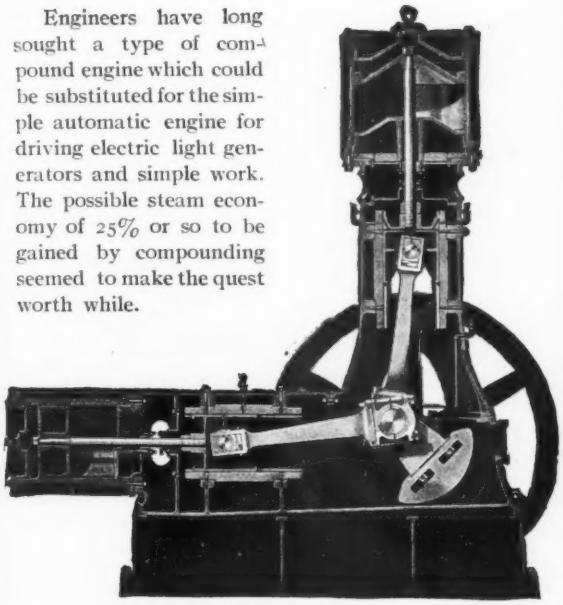
Sales Offices in all large cities

A PERFECT COMPOUND ENGINE

Engineers have long sought a type of compound engine which could be substituted for the simple automatic engine for driving electric light generators and simple work. The possible steam economy of 25% or so to be gained by compounding seemed to make the quest worth while.



However, most of the compound engines proposed for this work have hitherto had shortcomings which overbalanced, at least in part, the advantage of smaller steam consumption.

Tandem compounds occupy too much room and require large foundations. There are no advantages in the way of more even turning moment or better balancing.

Cross compounds meet with the same objections, but in a more aggravated form, as to expense for foundations and space; further the weight and cost of base castings is practically doubled. Cross compounds are no easier to balance than are simple engines.

The ideal solution of the problem has been arrived at, however, in the AMERICAN BALL ANGLE COMPOUND ENGINE. The floor space occupied is no larger than would be required for a simple automatic engine of the same power. Placing two cylinders at right angles gives accurate balancing in all directions, which is impossible to exactly balance the reciprocating single cylinder engine without introducing an unbalanced condition in a direction at right angles to the piston travel, but in the ANGLE COMPOUND the balancing is perfect. Two cylinders at right angles give a more even turning moment, and because of this there is less friction upon the bearings and pins, and alternators driven by ANGLE COMPOUNDS will run parallel without any trouble and without use of heavy wheels.

If you are in any way interested in engine performance, it will be worth your while to send for our handsomely illustrated Bulletin No. 14, explaining all details of this remarkable engine.

AMERICAN ENGINE CO.,
22 Raritan Ave., Bound Brook, N. J.

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20 x 42 Allis, Corliss	16 x 32 Buckeye, Automatic
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Also large assortment of Automatic and Throttling Engines of other sizes.

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84 x 18, 78 x 16, 72 x 18 High Pressure, 72 x 16, 66 x 16 and smaller sizes of Horizontal Tubular Boilers, also Vertical and Fire Box Boilers from 5 to 60 H. P.

Heaters, Pumps, Tanks, Saw Mills, Iron Working Tools, etc., etc.

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We are the sole manufacturers of the celebrated "Leader" Injectors and Jet Pumps.

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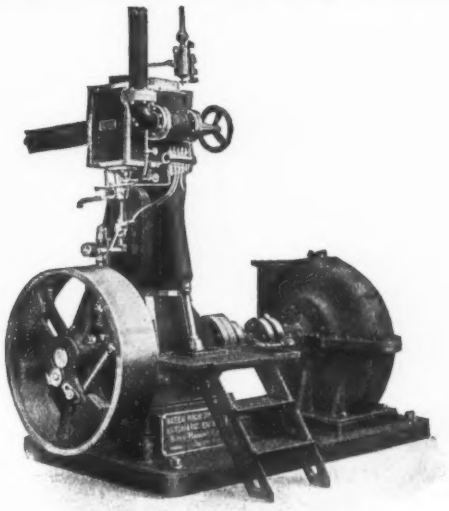

Bates Vertical Automatic High Speed Engine for connection to Pumps. Especially adapted for

WATER WORKS, IRRIGATION

and other places where it is desired to move a large volume of water at small expense.

CATALOG FOR THE ASKING.

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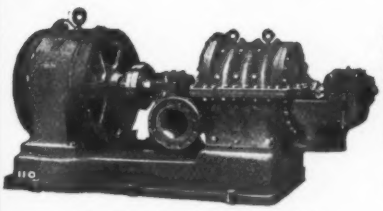

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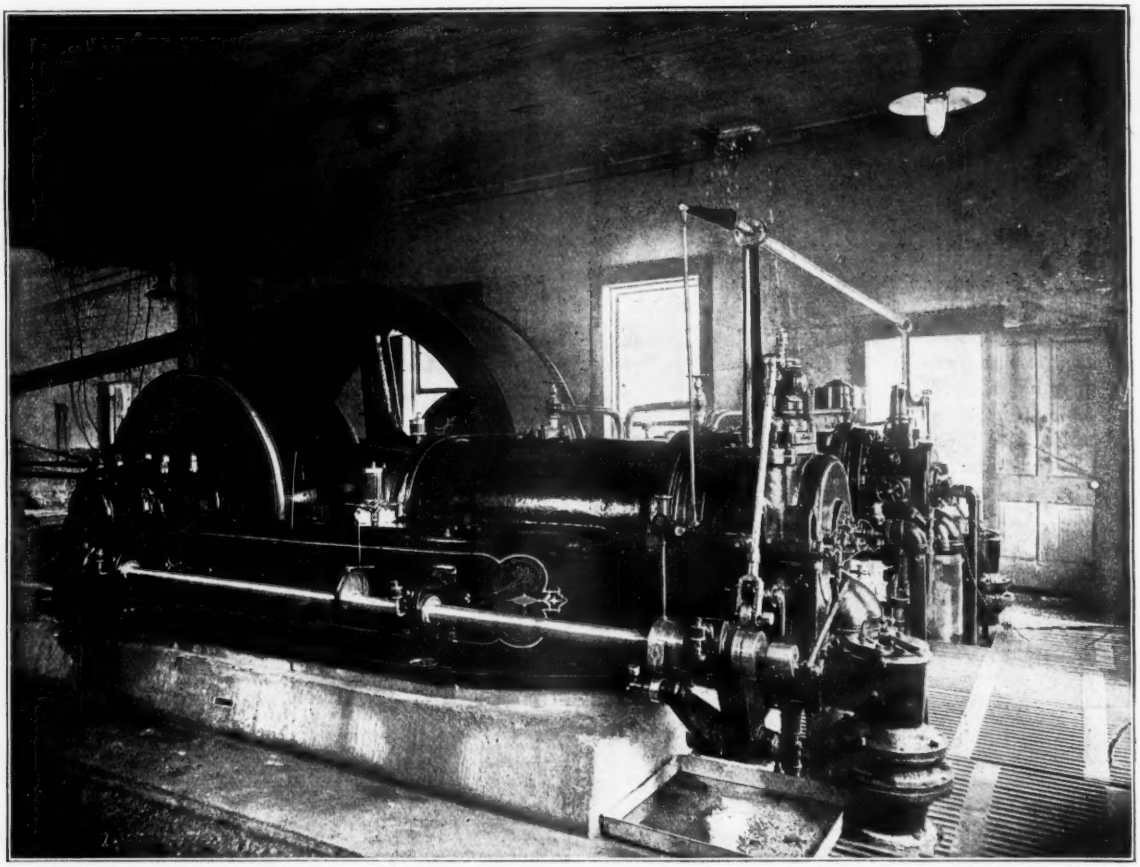

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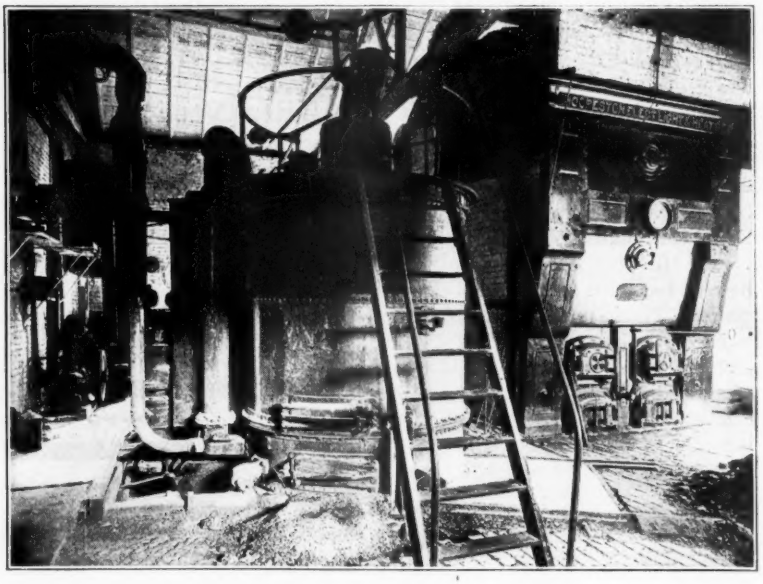

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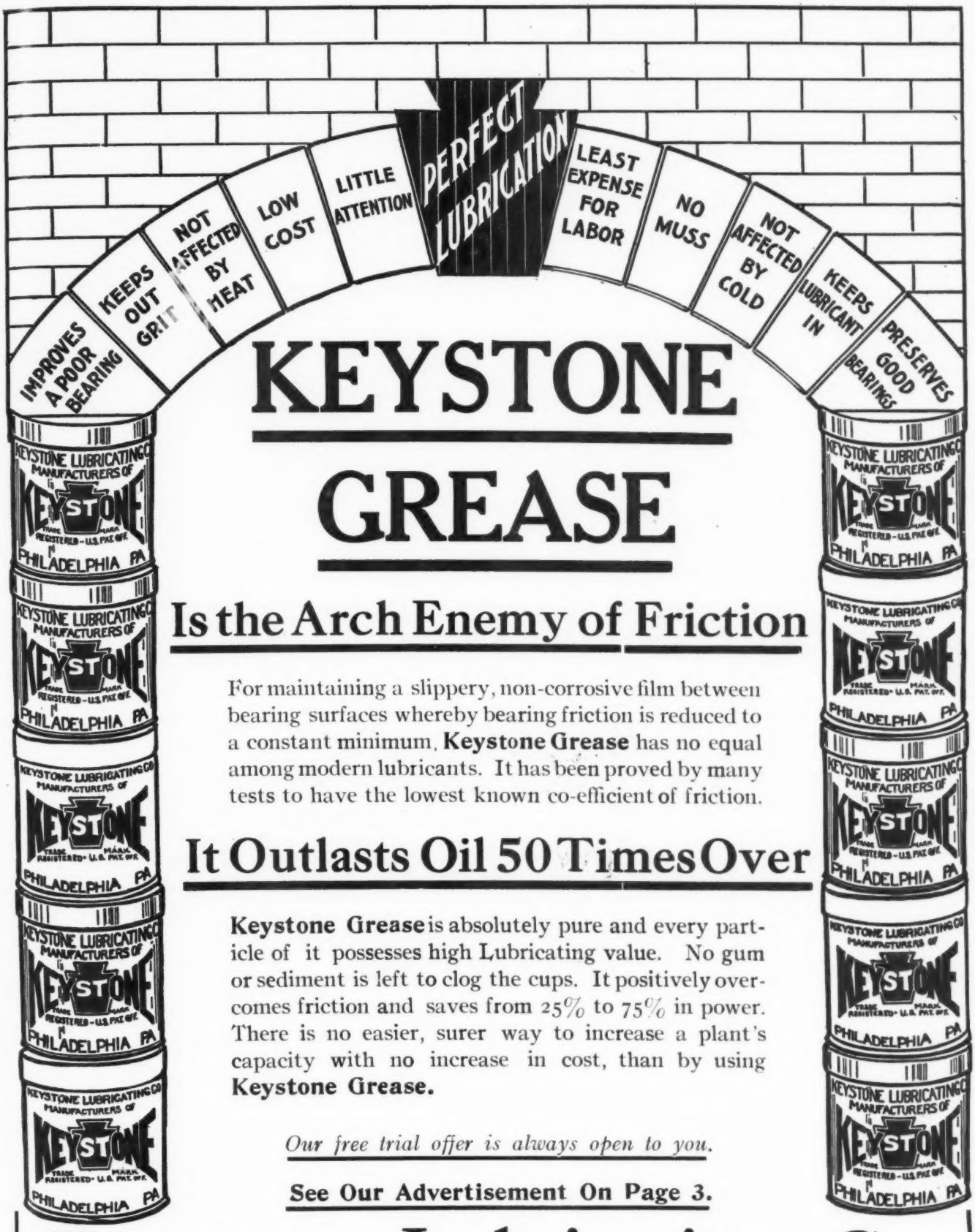

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